NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

TECHNICAL NOTE 3110

TRENDS OF ROLLING-CONTACT BEARINGS AS APPLIED TO AIRCRAFT GAS-TURBINE ENGINES

By Panel on High-Speed Rolling-Contact Bearings

SAE Summer Meeting, Atlantic City, N. J.

Washington
April 1954

FOREWORD

In view of the current interest in the subject, the Society of Automotive Engineers organized a panel session on "HIGH-SPEED ROLLING-CONTACT BEARINGS" at the Society's 1952 Summer Meeting. The members of the Panel and their affiliations at that time were: Edmond E. Bisson, NACA, Cleveland, Ohio; Stephen Drabek, General Electric Co., Cincinnati, Ohio; Daniel Gurney, Marlin-Rockwell Corp., Jamestown, N. Y.; E. Fred Macks, NACA, Cleveland, Ohio; F. W. Wellons, SKF Industries, Philadelphia, Penna.; C. M. Michaels, Wright Air Development Center, Dayton, Ohio.

Since all the Panel members, except Mr. Drabek, were either members of the NACA Subcommittee on Lubrication and Wear or on the staff of the NACA Lewis Laboratory, it was considered desirable for the NACA to assemble and release the papers which were presented, since the SAE did not plan to publish this material in full.

The general discussion of "trends of rolling-contact bearings as applied to aircraft turbine engines," which appears on the yellow pages in this report, was prepared at the Lewis Laboratory as a general summary of the discussion at the meeting and it was subsequently coordinated with the members of the Panel to insure its accuracy.

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SUMMARY

A review of the present and future trends of rolling-contact bearings for aircraft gas-turbine engines indicates that while rolling-contact bearings are satisfactory for present production engines, such bearings for future, higher output aircraft engines require extensive development. The specific requirements for rolling-contact bearings for future aircraft gas-turbine engines are very severe. Operating temperatures to 750° F, turbine soak-back temperatures to 1000° F, thrust loads to 50,000 pounds, and DN values to 3.5×10^6 are desired, if possible of attainment. (DN value is the product of the bearing bore in millimeters and shaft speed in rpm.)

These operating conditions result in two primary problems:

(a) Cage materials

(b) Endurance of materials for raceways and rolling elements

Besides these specific requirements, there are the following basic requirements:

(1) Extreme reliability for a life of 1000 hours

(2) Insensitivity to oil flow interruptions

(3) Low oil flow and cooling load

(4) Freedom in alinement tolerances

(5) Low starting torque without preoiling

(6) Ability to start at very low temperatures

It is generally believed that substantial bearing improvements can be made, particularly if the efforts of bearing companies, engine companies, and government agencies are increased and integrated where possible.

INTRODUCTION

In view of the increasing problems associated with high-speed, high-temperature bearings for aircraft, a panel discussion on "High-Speed Rolling-Contact Bearings" was held during the SAE summer meeting, June 4, 1952. The purpose of this report is to preserve the complete text and figures of the six prepared papers presented during the panel discussion, and these are given in appendixes A through F. The text to follow is based on the original papers and the discussion at the meeting.

DISCUSSION

The prepared papers and the discussion which followed have been analyzed and summarized under the following headings:

(a) Requirements

(b) Approaches to the Problem

REQUIREMENTS

The rolling-contact bearing is favored over the hydrodynamic bearing in the subject application because of:

(a) Extreme low-temperature starting

(b) Low starting torque without preoiling

(c) Relative insensitivity to oil flow interruption

(d) Greater freedom in alinement tolerances

(e) Lower oil flow and cooling load

The foregoing are regarded as basic requirements for aircraft gas-turbine engines and are in addition to the requirements to follow.

It is generally agreed that rolling-contact bearings are satisfactory for present production aircraft gas-turbine engines. The lack of acute problems encountered in these present engines does not mean that no problems have existed. Two primary difficulties which were associated with turbojet rotor bearings and which have been solved, at least for the present, were:

(a) Inadequate bearing lubrication and cooling provided by oil-air-mist lubrication which resulted in bearing failures. This difficulty was generally solved by oil-jet lubrication for all rotor bearings.

(b) High-temperature "soak-back" from the turbine wheel to the turbine rotor bearing which resulted in evaporation of the lubricant at the bearing location. The lack of lubrication during subsequent starts resulted in a high rate of bearing failures because of "pick-up" and metal transfer caused by incompatible cage and raceway materials. An interim solution to this type of failure was obtained, in large part, by silver plating the brass and bronze bearing retainers.

Although presently not causing any great concern, the foregoing two problems may be serious limitations for future engines. This is true because, as engine performance is extended, both bearing cooling and turbine soak-back temperature will become more critical to reliable bearing performance, and it is not unlikely that these problems may reoccur.

It is generally agreed that bearings for future higher output aircraft engines require extensive development. The direct requirements in this regard are not specific, since engine manufacturers have various projected designs, the exact requirements of which are quite different in many cases.

The bearing problem most common to all engine manufacturers is that of increasing bearing temperatures. Operating bearing temperatures of 750° F are considered representative in the foreseeable future. Bearing temperatures of 1000° F due to turbine soak-back have been discussed.

A second requirement of utmost importance is that of high bearing thrust loads. The bearing thrust load becomes a more severe problem as engine output is increased. Not only are the thrust loads very great, but the magnitude of this load varies widely over the range of engine operating conditions. Design values of bearing thrust load exceeding 50,000 pounds would undoubtedly be employed in some engine designs if bearings existed which would operate for the desired 1000-hour life under this load at the temperatures and speeds involved. Until bearings are developed to absorb such enormous loads, these loads must be reduced by the use of balancing pistons. The use of balancing pistons dictates the use of seals at large diameters, and the combination of balancing pistons and large seals results in a substantial loss of engine performance and an increase in complication and weight. A third requirement of importance to many engine manufacturers is that of higher bearing surface speeds. Although rotor rotative speeds are not increasing significantly, very large bearings must be used with dual-rotor engines. Bearing surface speeds of 3.5×10^6 DN value have been discussed as quite probable if reliable bearings were available. This requirement is about three times the surface speeds used in present-day aircraft turbine engines.

One of the very important problems in present and future turbine engine bearings is that of the cage material. Incompatibility of cage materials with race materials has resulted in bearing failures in the past and will undoubtedly be a serious limitation on future bearing performance because the lubrication of the cage-locating surface will certainly be poorer at the elevated speeds and temperatures. This limitation on bearing performance will continue to exist until considerable research information is obtained on compatibility of materials which have adequate physical properties at temperatures to 750° F. This information must be obtained in full-size bearings as well as in bench tests on friction and wear.

As a result of the foregoing bearing requirements, several additional problems result, foremost of these being:

(a) Lubricants for higher temperatures

(b) Greatly increased lubricant cooling load

As an example of lubricant cooling load due only to bearing losses, it is estimated that about 8.5 horsepower is rejected to the oil in one type of production aircraft gas-turbine engine. Some of the engines in the development stage have lubricant cooling loads many times the foregoing value. To this cooling load must be added the large heat loss to the bearing (and subsequently to the lubricant) from the high-temperature environment.

Although the two problems of high-temperature lubricants and lubricant cooling cannot be separated from the high-temperature, high-load, high-speed bearing problem, the scope of the present report does not include a detailed lubricant discussion. A discussion of lubricants may be found in reference 1.

The anticipated operating conditions of rolling-contact bearings in turbojet engines may be summarized as follows:

Maximum bearing temperature, °F	
Operating	750
Turbine soak-back	1000
Maximum bearing thrust load, lb	50,000
Maximum bearing surface speed, DN value	3.5×10^6

APPROACHES TO THE PROBLEM

The various bearing requirements (such as increase of temperature, thrust load, and surface speed) cannot be considered independently. Increased engine performance results in an increase in all three of these conditions, with bearing surface speed being a possible exception in certain cases only.

General

The solution of the problems associated with adequate bearings for future aircraft engines is the responsibility of:

(a) The research engineer and the physicist

(b) The metallurgist

(c) The bearing application engineer

There are many possibilities toward the extension of the temperature, load, and speed limitations of existing bearings. Materials and design are the two primary avenues of approach.

Materials for Races and Rolling Elements

Physical properties. - The required physical properties of materials suitable for races and rolling elements of high-speed, high-temperature bearings are as follows:

(1) Minimum hardness of Rockwell C-58 at operating temperature

(2) Dimensional stability at operating temperature

(3) Critical alloying elements (particularly tungsten) held to a reasonable minimum

(4) Oxidation resistance at operating and at room temperatures

(5) Reasonable heat-treatment and grinding characteristics

(6) Readily available from several sources

Of the various possibilities, the molybdenum tool steels appear to show considerable promise toward meeting these physical properties. Hot-hardness tests show that some of these steels will meet the minimum hardness value of Rockwell C-58 at a temperature of 800° F; hardness drops to Rockwell C-57 at 900° F. Dimensional stability studies to 900° F showed negligible changes in diameter of rings made from these steels. These steels do, however, require considerably longer grinding times (80 to 300 percent) than does SAE 52100.

Compatibility with cage material. - The problem of compatibility of those rolling-contact bearing materials which are in rubbing contact is quite severe and is believed to be one of the present limitations to high-speed operation. This problem will be discussed more fully in the section "Cage Materials" under "Compatibility."

Fatigue. - Provided other limitations can be eliminated, fatigue will become the limitation for the thrust bearing operating at high loads. The fatigue life of rolling-contact thrust bearings should therefore be increased significantly if this bearing type is to be a reliable component of future aircraft engines. For example, as based upon existing knowledge, the life of a ball bearing is limited to 1000 hours with 10 percent expected failures at zero external load if DN values of the order of magnitude of 2×10^6 are employed. The bearing life is limited because of internal centrifugal loads. To solve this problem, extensive research on materials must be conducted by bearing and engine companies as well as by government agencies.

While some feel that the future aircraft engine requirements for rolling-contact bearings are in the "realm of fancy," many others think very significant improvements in bearing fatigue life can be made. Tools are now available (ref. 2) which show promise toward accelerating this type of research. Bench tests must, however, be followed by full-scale bearing tests. The problem is as difficult as it is important. It is believed that the effort expended in this field must be greatly increased in order to improve the fatigue life and to decrease the life dispersion of bearings.

Cage Materials

Physical properties. - The required physical properties of materials suitable for cages of high-speed, high-temperature rolling-contact bearings are as follows:

(1) Adequate strength at low and high temperatures (present cage materials have adequate strength at low temperatures)

(2) Oxidation resistance

(3) Thermal expansion coefficient approaching that for the race materials

Besides these physical property requirements, the property of compatibility is most important; this property will be discussed in the following section.

Compatibility. - The compatibility of those materials in rolling contact bearings which are in rubbing contact must be improved as the temperatures, loads, and speeds are increased. Experimental research must be accelerated, and conditions simulating actual conditions in bearings should be used. As certain of these conditions are not known exactly, the range of variables in bench tests must be increased to cover the unknown region. Bench tests must be followed by full-scale bearing tests under simulated operating conditions. In addition to the conventional friction and wear data, fundamental research regarding factors such as solid solubility may yield helpful results. The effect of solid surface films resulting from internal and external sources (as discussed in detail in ref. 3) as well as coatings and platings should also be investigated further, as many feel there are too little data available in this field. The ability of materials to form surface films that prevent welding is an important factor in both dry friction and boundary lubrication.

Design

Some improvement can be expected through geometric changes in cages as well as internal bearing design. Cage-locating and cage-pocket surfaces should be provided with lubrication as near to hydrodynamic (full film) as is possible. Balanced, light cages might show some improvement. Cage surface loads may be reduced appreciably by adequate internal bearing design. Reduction of these loads would lessen the severity of the compatibility problem. Refinement of tolerances (for rolling element uniformity and roundness and for the precision of raceway tracks) must be made as operating conditions become more severe. Experimental determination of optimum combinations of bearing diametral and axial clearances and contact angle for particular operating conditions may also yield substantial increases in bearing performance. Undercuts and chamfers may also be extremely significant as bearing limitations are extended. Surface finish tolerances and dimensional stabilization at the operating conditions are also items to be improved. Future bearing lubrication may be made more effective by means of precisely controlled lubricant flows over and within the bearings.

CONCLUDING REMARKS

The requirements for rolling contact bearings for future aircraft gas-turbine engines are very severe. Operating temperatures to 750° F, thrust loads to 50,000 pounds, and DN values to 3.5×10^6 are desired.

RECOMMENDATIONS

Specific recommendations are given in the text. It is believed that substantial bearing improvements can be made if the efforts of bearing companies, engine companies, and government agencies are increased and integrated.

Lewis Flight Propulsion Laboratory
National Advisory Committee for Aeronautics
Cleveland, Ohio, December 16, 1953

REFERENCES

1. NACA Subcommittee on Lubrication and Wear: Review of Current and Anticipated Lubricant Problems in Turbojet Engines. NACA RM 51D20, 1951.

2. Macks, E. F.: The Fatigue Spin Rig - A New Apparatus for Rapidly Evaluating Materials and Lubricants for Rolling Contact. Lubrication Eng., vol. 9, no. 5, Oct. 1953, pp. 254-258.

3. Johnson, Robert L., Swikert, Max A., and Bisson, Edmond E.: Investigation of Wear and Friction Properties Under Sliding Conditions of Some Materials Suitable for Cages of Rolling-Contact Bearings. NACA Rep. 1062, 1952. (Supersedes NACA TN 2384.)

APPENDIX A

PROBLEMS PERTAINING TO HIGH-SPEED ROLLING-CONTACT AIRCRAFT BEARINGS OF CONCERN TO THE BEARING INDUSTRY

By Daniel Gurney, Vice President
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Bearings in which the load is carried by means of balls or rollers contacting races are becoming universally known as rolling-contact bearings. They are entirely different in operating principal from hydrodynamic or oil-film bearings in which the load is supported by a fluid film. In the immediate past, ball and roller bearings have been called antifriction bearings. Their friction is relatively low, particularly at starting. However, hydrodynamic bearings may also have low friction and their designers certainly would object to their being called "friction bearings." Thus rolling-contact bearings is a much better descriptive name for this type of bearing.

Rolling-contact bearings may be broadly classified as ball bearings and roller bearings. Each of these classes is subdivided into bearings of different designs for different purposes. The deep-groove annular ball bearing consists of an inner and an outer grooved race between which is a complement of balls held in spaced relation by a suitable cage or retainer. This bearing is assembled by displacing the inner race radially within the outer race and filling the open side with slightly over 50 percent of the full complement of balls. The balls are then equally spaced and the cage is assembled to hold them in this spaced relation. This bearing does not have the maximum number of balls, therefore it does not have the maximum load carrying capacity.

The maximum-capacity ball bearing is sometimes made by modifying the deep groove bearing by using two filling slots to introduce the last few balls that cannot be put in by eccentric displacement of the races. This is known as the notched-type bearing.

The counterbored type of bearing is one having the outer race groove shallow on one side. This bearing is made with the maximum possible ball complement, but is capable of taking thrust in only one direction. The depth of the groove on the counterbore or shallow side is equal to the amount that the outer ring can be expanded by heating it about 300° F above the temperature of the inner ring and balls, and the bearing is assembled by so heating it and snapping it over the inner race, balls, and cage.

The angular-contact bearing is a variation of the counterbored type of bearing in which the balls contact the race grooves at a predetermined angle which makes the bearing capable of taking a heavier thrust load. This angle is governed solely by the looseness of the internal "fit-up" of the bearing and the conformity of the race-groove curvature to the ball curvature.

All the aforementioned bearings may be made in double-row types where two rows of balls are used with single inner or outer rings, or both.

Roller bearings also are made in many different designs or classes which will be briefly enumerated. Cylindrical roller bearings consist of an inner and an outer race and rollers spaced by a cage. Usually the rollers have approximately the same length as diameter, although for relatively low speed and high-load capacity they may have long rollers. However, the longer the roller is in proportion to its length, the more difficult it becomes to prevent it from skewing.

Tapered roller bearings have a tapered inner and outer race or cone and cup, respectively, between which are mounted tapered rollers suitably spaced by a cage. The longitudinal elements of all of these tapers meet at one point on the center line of the bearing so there is virtually true rolling action of the elements.

Rolling-contact bearings carry their load at the contact of the rolling elements with the raceways. This results in relatively high stresses. The life of the bearing is limited by the fatigue strength of the metal. The races and rolling elements are both subjected to a number of stress cycles per revolution. Thus the life of the elements is governed by the number and intensity of these stress cycles.

The life of a series of bearings is inversely proportional to the third power of the load. Thus, if the load is cut in half the bearing will last eight times as long.

Unfortunately, the predicted life of bearings is based on the assumption that the bearing fails as a result of fatigue of the races and/or of the rolling elements at the rolling contacts. For bearings used on the main shafts of aircraft gas turbines, this is not usually the case. A great many of the failures of turbine main shaft bearings are due not to the rolling contacts as such, but to the hydrodynamic bearings that must be used to control the rolling elements, in other words, the cage. Also, some of the failures are due to the fact that all the contacts are not purely rolling contacts. The ball tracks in the race grooves of angular-contact bearings often show three pitted bands separated by two shiny bands where true rolling occurred. There is and must be slippage in an angular-contact ball bearing under heavy thrust load. Therefore the ball contacts must receive lubrication.

Roller bearings also are never pure rolling-contact bearings. Every manufacturer does his utmost to make them as near to pure rolling as possible, but again the rolling elements have to be controlled by mechanisms which depend upon hydrodynamic bearings for their operation. The flanges which guide the rollers do so through sliding friction for which lubrication must be provided.

Perhaps the most troublesome member of both roller and ball bearings used in gas turbines is the cage used to space the rolling elements. It in turn is controlled (or located) as a crude plain bearing usually on either the outside lands of the inner race or the inside lands of the outer race. This is inherently a poor hydrodynamic bearing. Its length to diameter ratio is of the order of 1/30, and its clearance for film thickness is several times what a good plain bearing should have. It is true enough that this plain bearing is very lightly loaded; nevertheless, it is responsible for a high proportion of jet engine bearing trouble. A great deal more thought and research should be expended in solving this phase of the rolling-contact bearing problem.

There has been considerable improvement in performance brought about by the introduction of silver plate on the bronze. The use of modified H monel metal also helps, but it is very difficult to fabricate.

Another approach is to cut down the clearance at the cage support surface, but this can be done only by finding a material which has more nearly the same coefficient of expansion as does the raceway steel.

Figures 1 through 3 show various rotor mountings of roller bearings in a jet engine. Figure 4 shows a detailed drawing of a high-speed roller bearing.

One of the most successful cage materials for high-speed ball bearings is a phenolic as used on one of the jet engine thrust bearings of figure 5. Figure 6 shows the manner in which it is mounted in the compressor thrust position of the engine. The cage is machined from compound made in the form of tubing by winding fine cotton cloth on a mandrel and impregnating it with bakelite. The result is that there are absorbent cotton fibers at all the machined surfaces which hold enough oil for the cage to operate satisfactorily under marginal lubrication. Another advantage is that it is very light. The use of phenolic cages is limited, however, to temperatures of 300° F and below, which rules them out for future applications.

The problems faced in providing bearings for larger jet engines, with their ever-increasing shaft sizes and load, are difficult enough without at the same time being faced with higher operating and soak-back temperatures. Steels can, however, be developed to meet these ever-increasing temperatures. The main problem is to develop satisfactory cages and lubricants.

Roller bearings can be made to take the loads and speeds of the larger engines of the future, but the ball thrust bearings have been pushed to about their limiting speeds and loads.

Figure 7 shows two angular-contact double thrust bearings duplex ground so that they divide the thrust load in one direction, with one bearing usually taking the reverse thrust. This bearing has a thrust capacity almost double that of a single bearing.

Figure 8 is a very successful compressor thrust bearing which is a double thrust angular-contact bearing on the right, duplex ground with an angular-contact bearing on the left. Both bearings have their outer race grooves in a common outer race which is spherical on the outside diameter for self-alining purposes. This bearing is equipped with phenolic cages since it is used with an ambient temperature which is not too high.

The development of thrust bearings to be run at higher DN values at higher thrust loads and at higher temperatures presents some very challenging problems. The problem of lubricants is equally demanding. Then to this is added the problem of operating these bearings for periods of time after the lubricant fails, and the problem becomes nearly insurmountable. Solving it should require great ingenuity and resources.

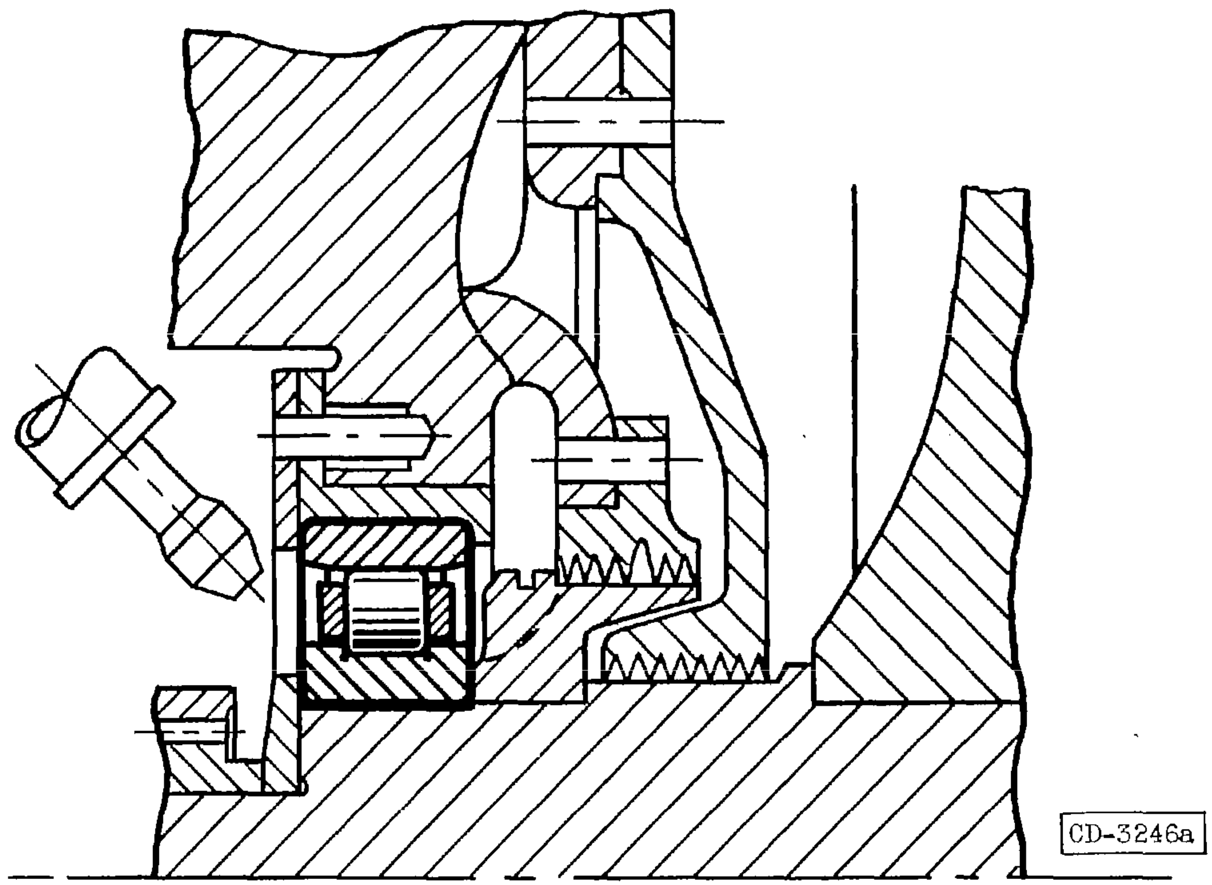

Figure 1. - Typical mounting of turbine engine forward bearing.

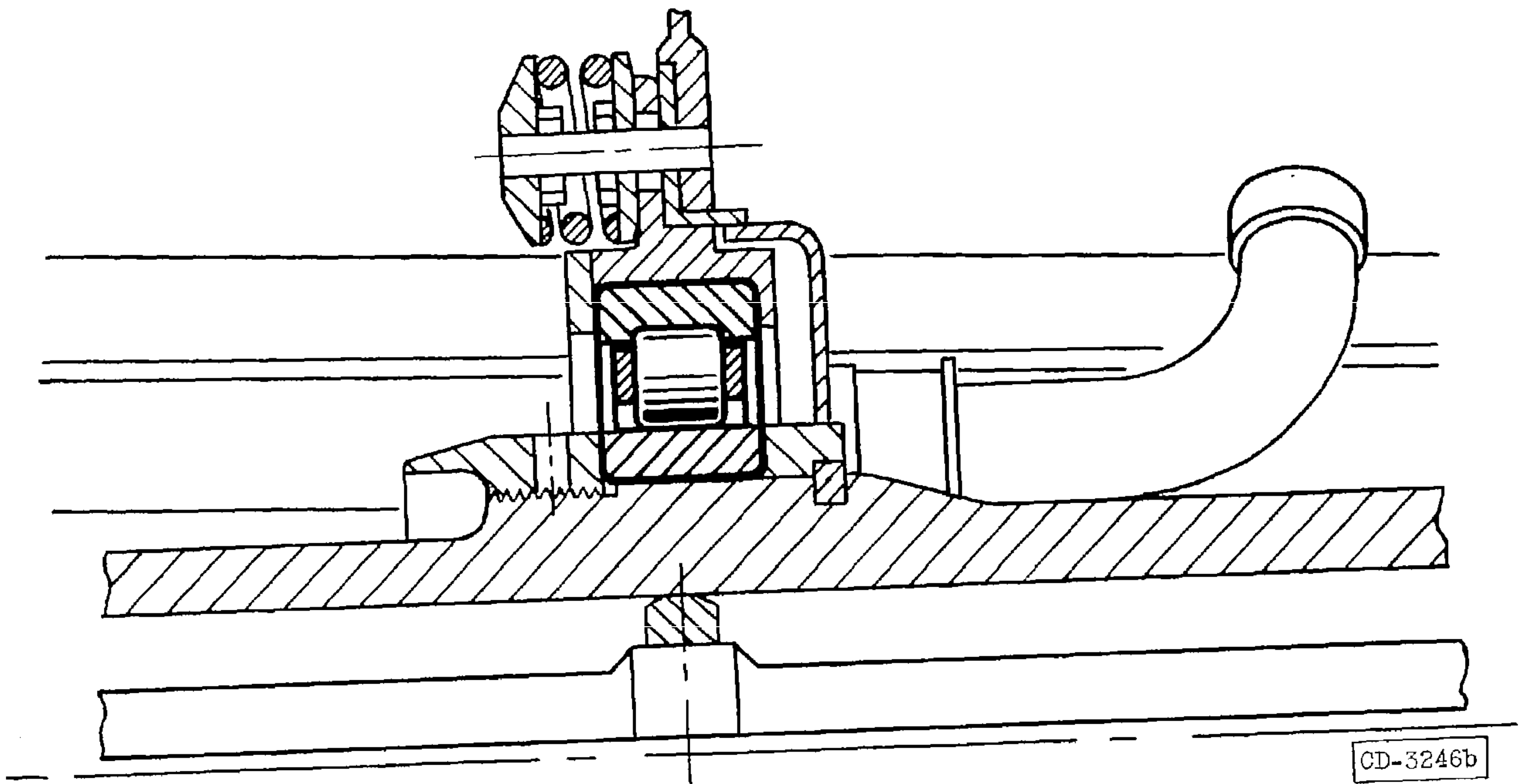

Figure 2. - Typical mounting of turbine engine damper (idler) bearing.

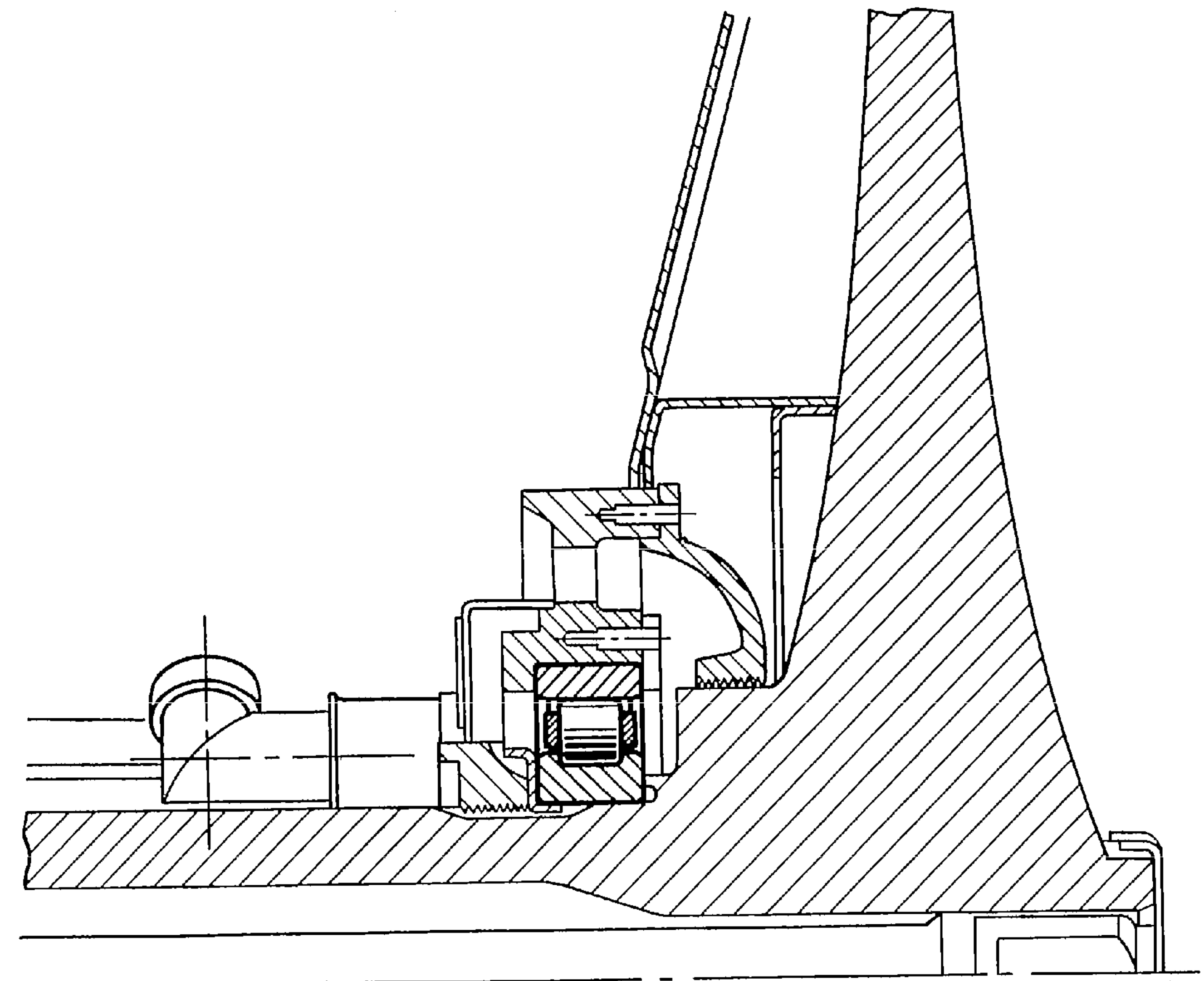

Figure 3. - Typical mounting of turbine engine rear bearing.

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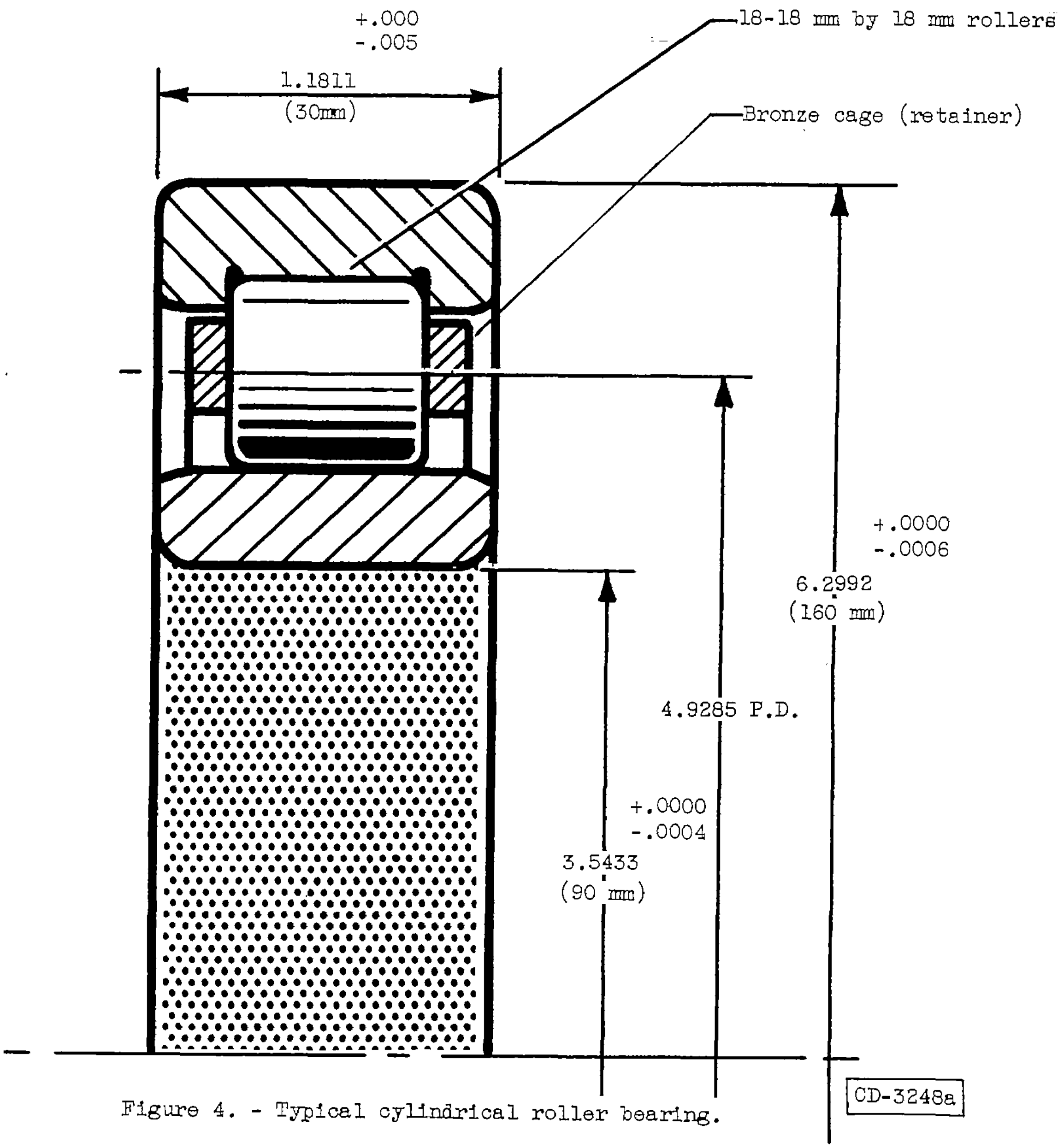

Figure 4. - Typical cylindrical roller bearing.

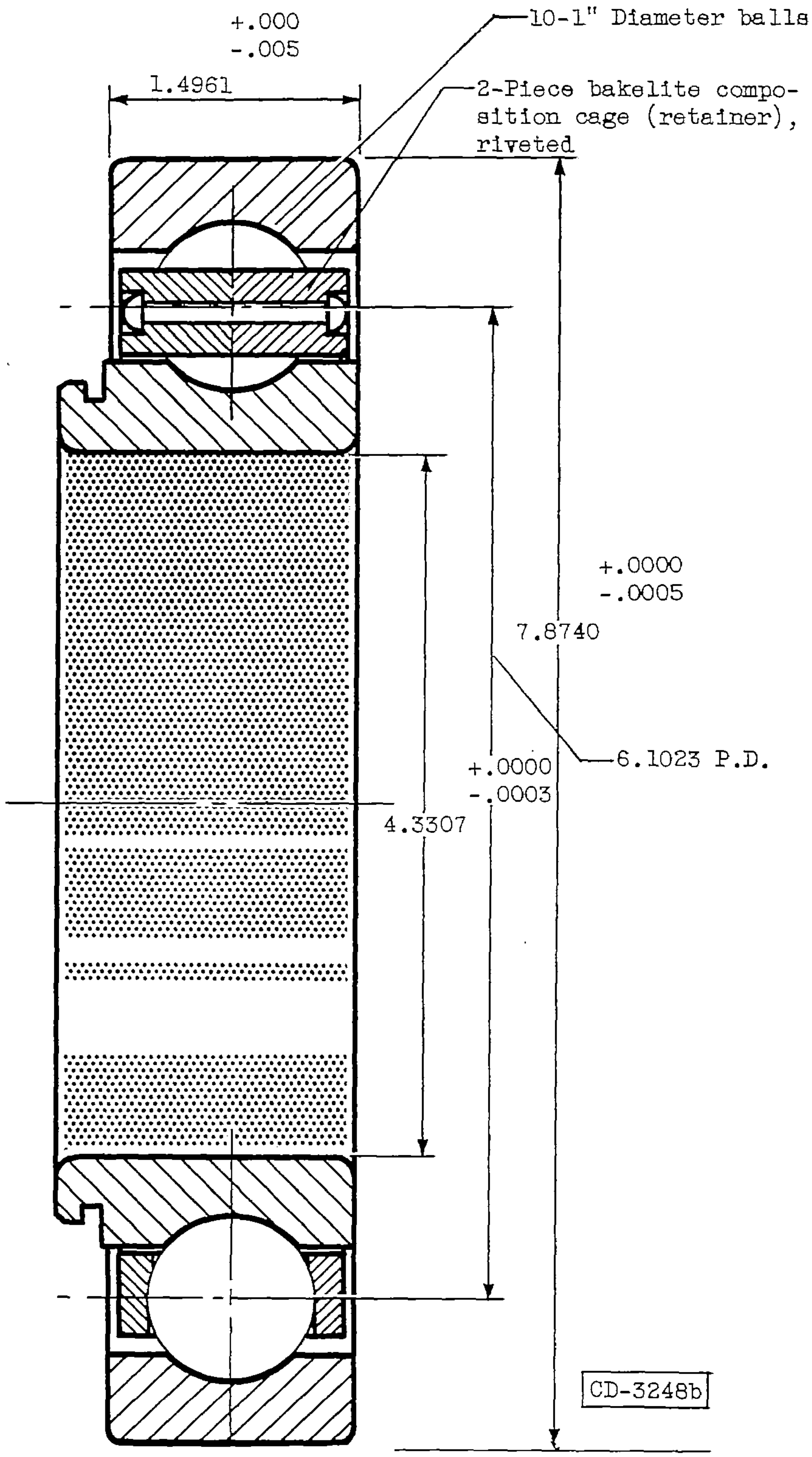

Figure 5. - Typical turbine engine thrust bearing.

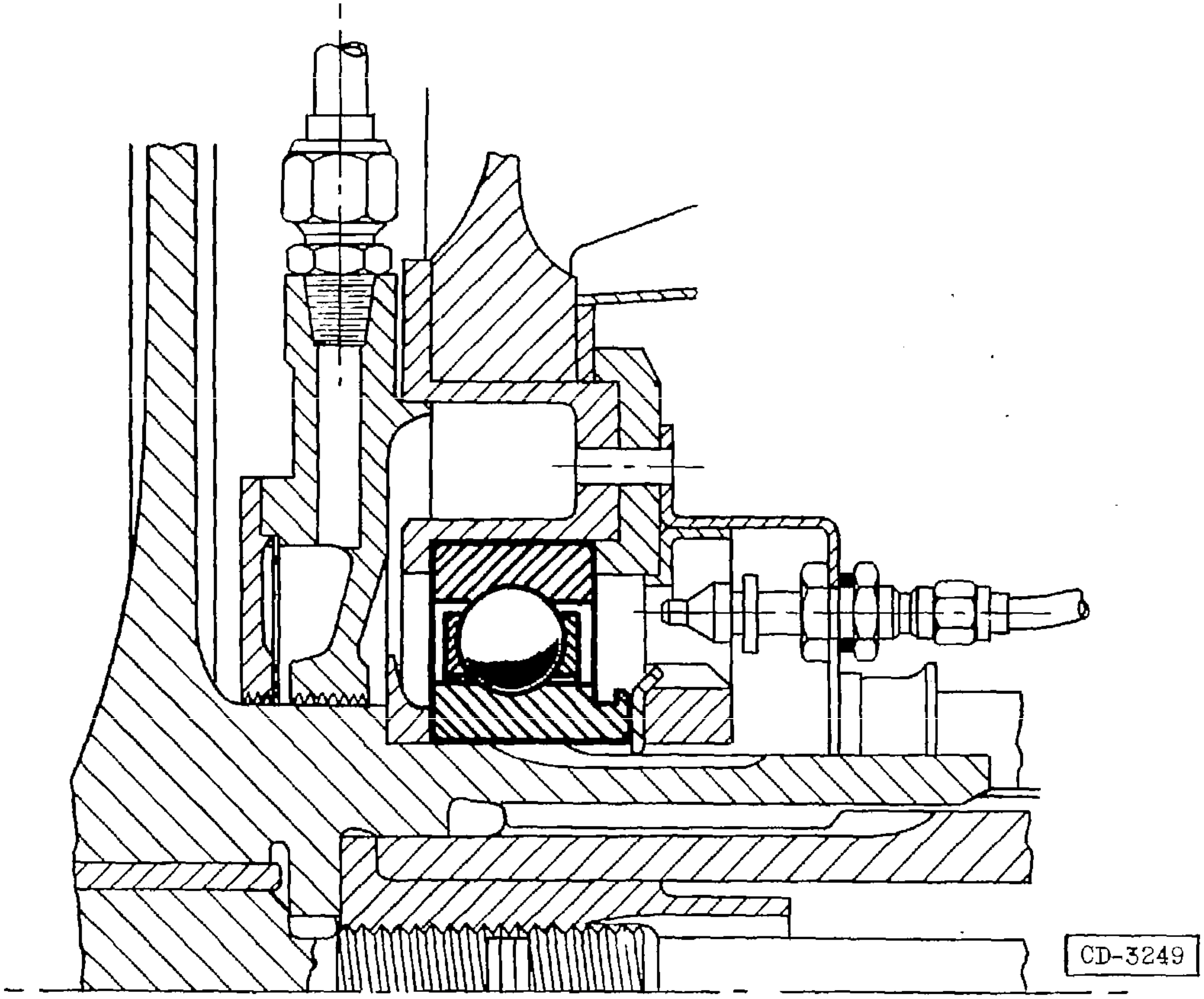

Figure 6. - Typical mounting of turbine engine thrust bearing.

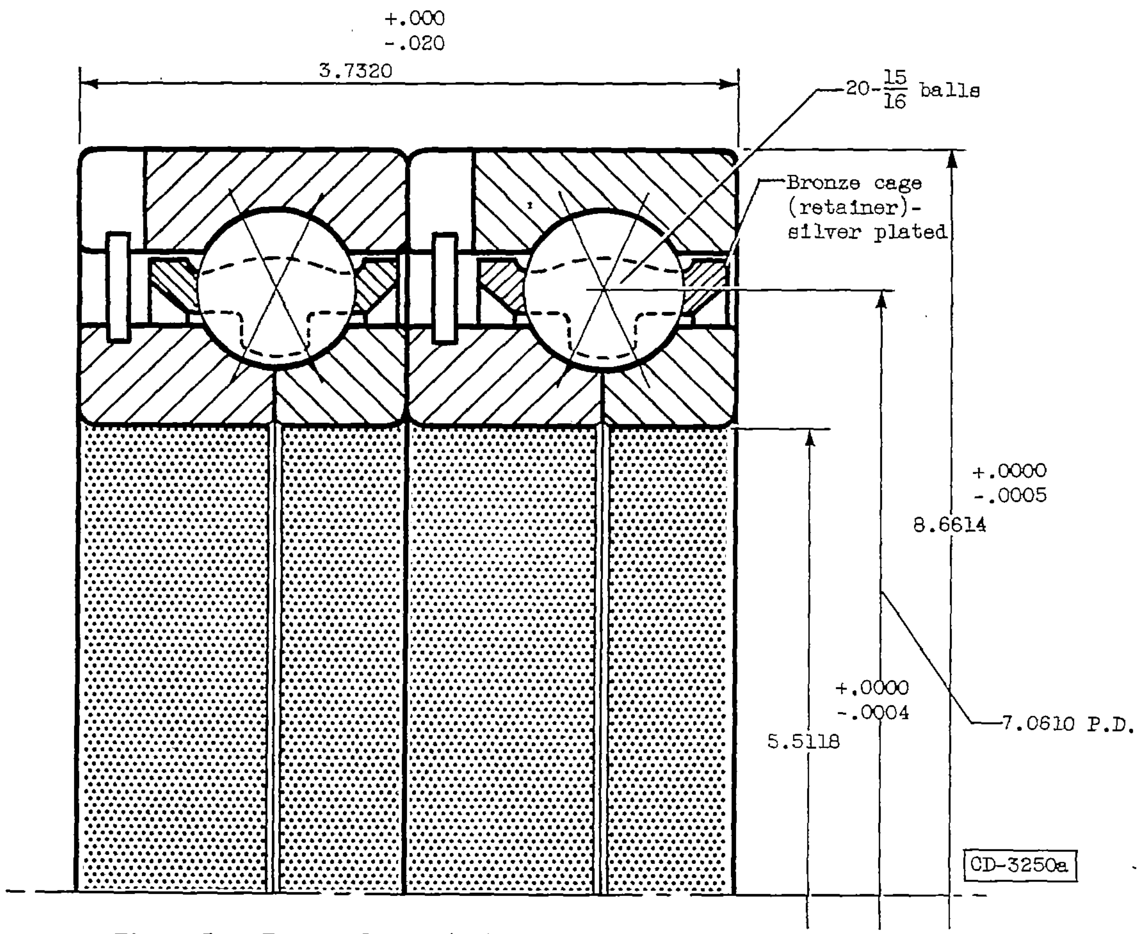

Figure 7. - Two angular-contact, double thrust bearings, duplex ground.

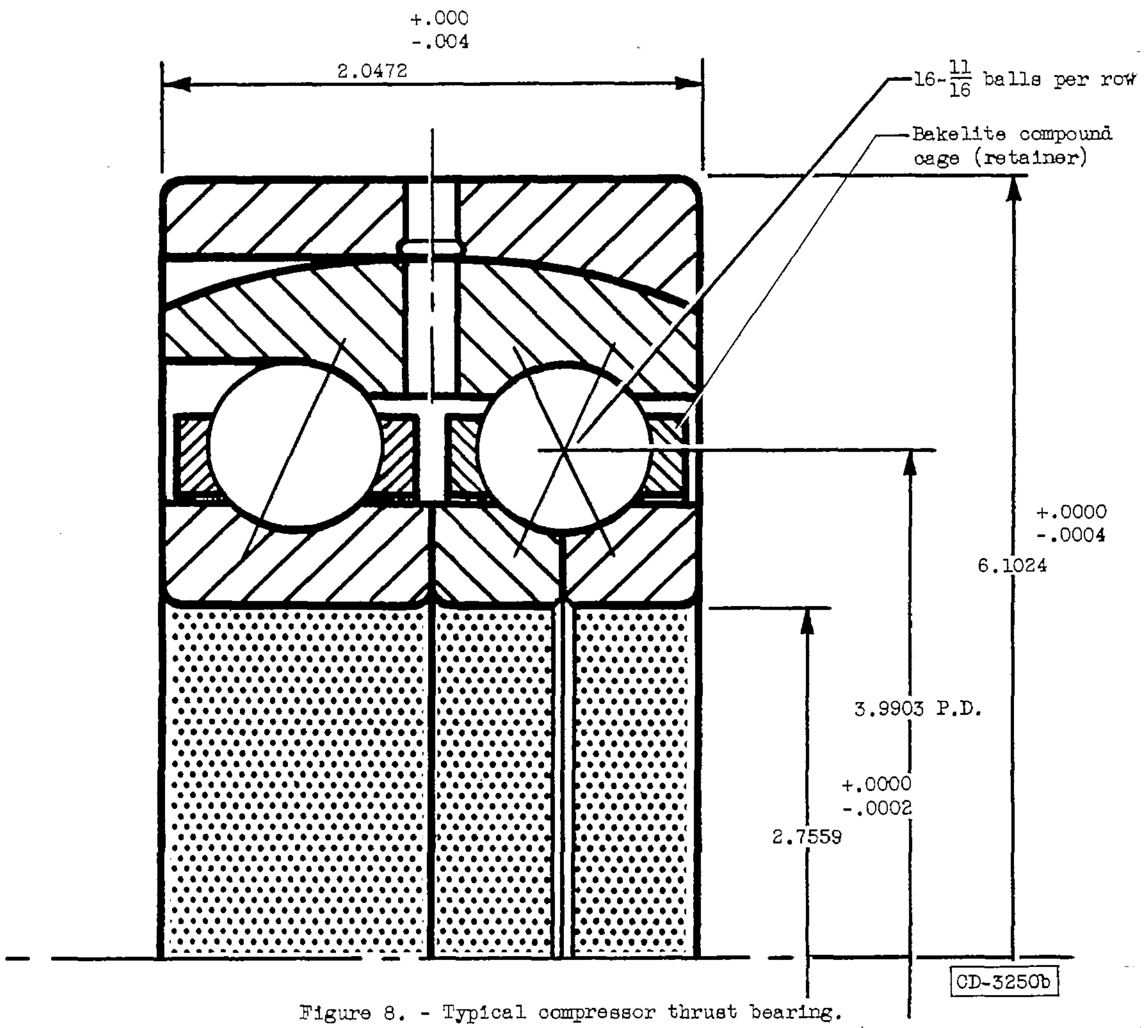

Figure 8. - Typical compressor thrust bearing.

APPENDIX B

PROBLEMS PERTAINING TO HIGH-SPEED ROLLING-CONTACT BEARINGS IN AIRCRAFT TURBINE ENGINES OF CONCERN TO THE MILITARY

By C. M. Michaels
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The mission of the Power Plant Laboratory of the WADC at Wright Field is to develop superior aeronautical power plant materiel conforming to USAF requirements and capable of standardization for use in a national emergency.

The military services, in accomplishing their mission, establish performance and test specifications for industry guidance in the development of equipment. Recognizing the need for aircraft power plants capable of operation at the extremes of temperature that could be encountered in the support of global offensive or defensive action, the industry has been requested to design power plants capable of fast start and operation within the ambient temperature ranges of -65° to 160° F.

Mr. Gurney, in his comments, emphasized the importance of the high-speed rolling-contact bearing in the design of the aircraft gas turbine. The importance of these bearings, their lubrication, and cooling cannot be overemphasized, particularly in view of the trend toward supersonic flight speeds. The demands for higher thrust power plants present challenging problems in the design of bearings, since it can be expected that higher temperatures, higher loads, and higher rotating speeds will be encountered.

Revisions are periodically made to the General Engine Specifications and Bulletins which present design criteria and recommended practices as guidance in the design of military aircraft engines. These specification changes result from experience gained by the services and industry in the production, operation, and maintenance of aircraft and engines. Korean experience has revealed that pilots and aircraft have been saved in combat operations because engines have continued to run for fairly long periods after complete loss of the oil supply. Further, two engine tests at the Power Plant Laboratory at Wright Field have confirmed that present types of turbojet engines will continue to run and develop thrust for periods of 17 to 26 minutes after complete loss of the oil supply. This feature, while not intentionally provided in the past, has prompted the services to consider it as a necessary attribute for a fully acceptable turbojet engine. A minimum operation of 15 minutes is desired following loss of oil supply, even though serious damage may have been suffered by the engine at the end of that period.

The aforementioned Power Plant Laboratory tests were run on two engines of approximately 5000-pound thrust starting with the following differences: (a) the first engine employed the solid-oil-stream, full-scavenge lubrication system, and (b) the second engine was equipped with the air-oil-mist type of lubrication.

The first engine, employing the solid-oil-stream lubrication system, operated for 17 minutes following oil supply cut-off. The vibration was normal until immediately preceding failure of the engine. An abrupt vibration increase was noted but failure occurred before the engine operator's throttle could be closed. Figures 1 through 4 show damage to this first engine.

The second engine, employing the air-oil-mist system, operated for 26 minutes following oil flow interruption. Normal vibration was experienced until suddenly a 12 mil vibration was noted. The engine was immediately shut down. Figures 5 and 6 show damage to this second engine.

Following these two tests, an engine contractor ran an additional test on an engine employing a solid-oil-stream lubrication system; this engine was provided with silver plated monel retainers in both ball and roller bearings. This engine operated without oil for a period of 52 minutes and was stopped when the damper bearing temperature reached 570° F. No damage to the engine resulted, but there was evidence of high-temperature operation at the damper and turbine bearings.

Those tests demonstrate the ability of antifriction bearings to operate for considerable periods without lubrication. It is recognized that test stand operation at static sea level does not simulate the high bearing loading encountered during flight maneuvers following loss of oil supply which could further appreciably shorten engine life.

The military services recognize the need for stimulating research and development effort on aircraft gas turbine bearings and their lubrication. To aid in this program, the Air Force is sponsoring, through the Coordinating Research Council, a joint project among the bearing, engine, and petroleum industries. For the present, an effort is being made to procure bearing test machines for industrial use. The first machine will be capable of testing bearings having

Bore, mm 90
Speed range, rpm . . . . 2000 to 20,000
Thrust load, lb max 15,000
Radial load, lb 10,000
Temperature soaking and control provisions for bearings and lubricant

The larger machine will test bearings having

Bore, mm 200
Speed range, rpm . . . . 500 to 12,000
Thrust load, lb max 100,000
Radial load, lb 25,000
Temperature soaking and control provisions for bearings and lubricant

In summary, the high-speed rolling-contact bearing, its design, metallurgy, and lubrication are recognized as key components in the design of aircraft gas turbines. There exists an urgent necessity to focus increased research and development effort on this subject to satisfy the ever-increasing demands on bearings insofar as load, temperature, and speed are concerned.

Figure 1. - Number 2 bearing failure of engine employing solid-oil-stream lubrication system showing absence of micarta retainer and bunching of balls.

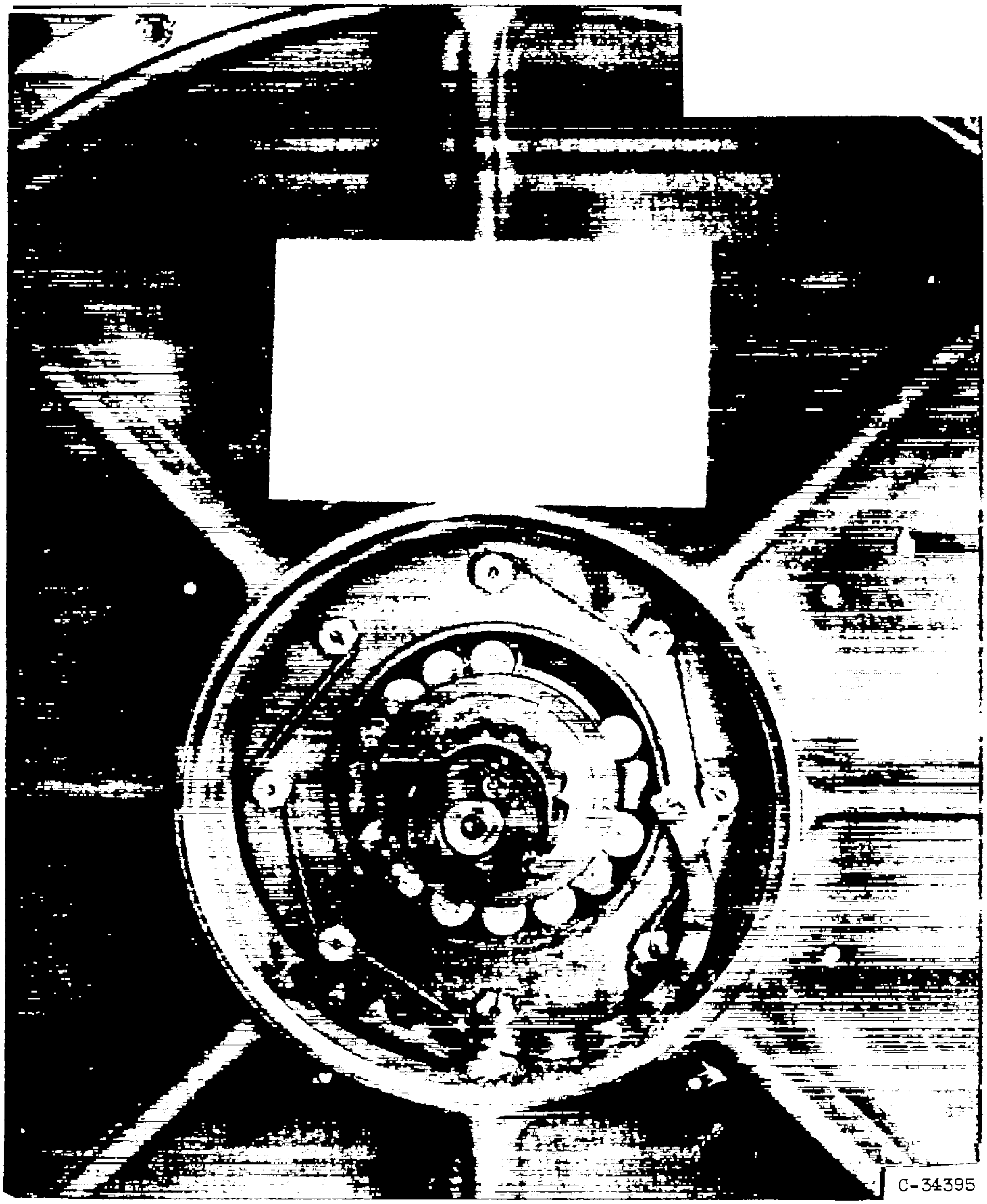

Figure 2. - Number 1 bearing failure of engine employing solid-oil-stream lubrication system showing failed retainer and rollers.

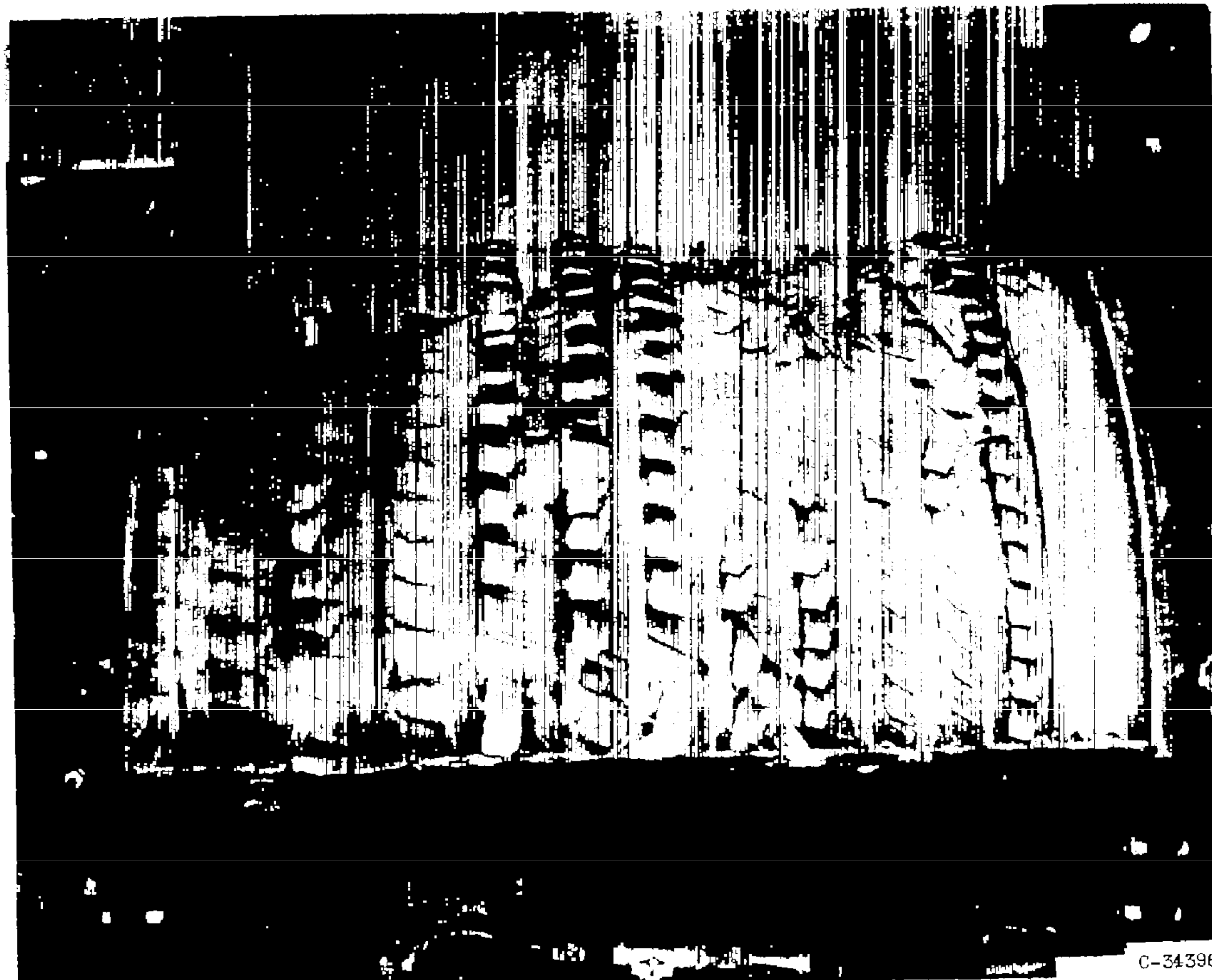

Figure 3. - Compressor damage of engine employing solid-oil-stream lubrication system.

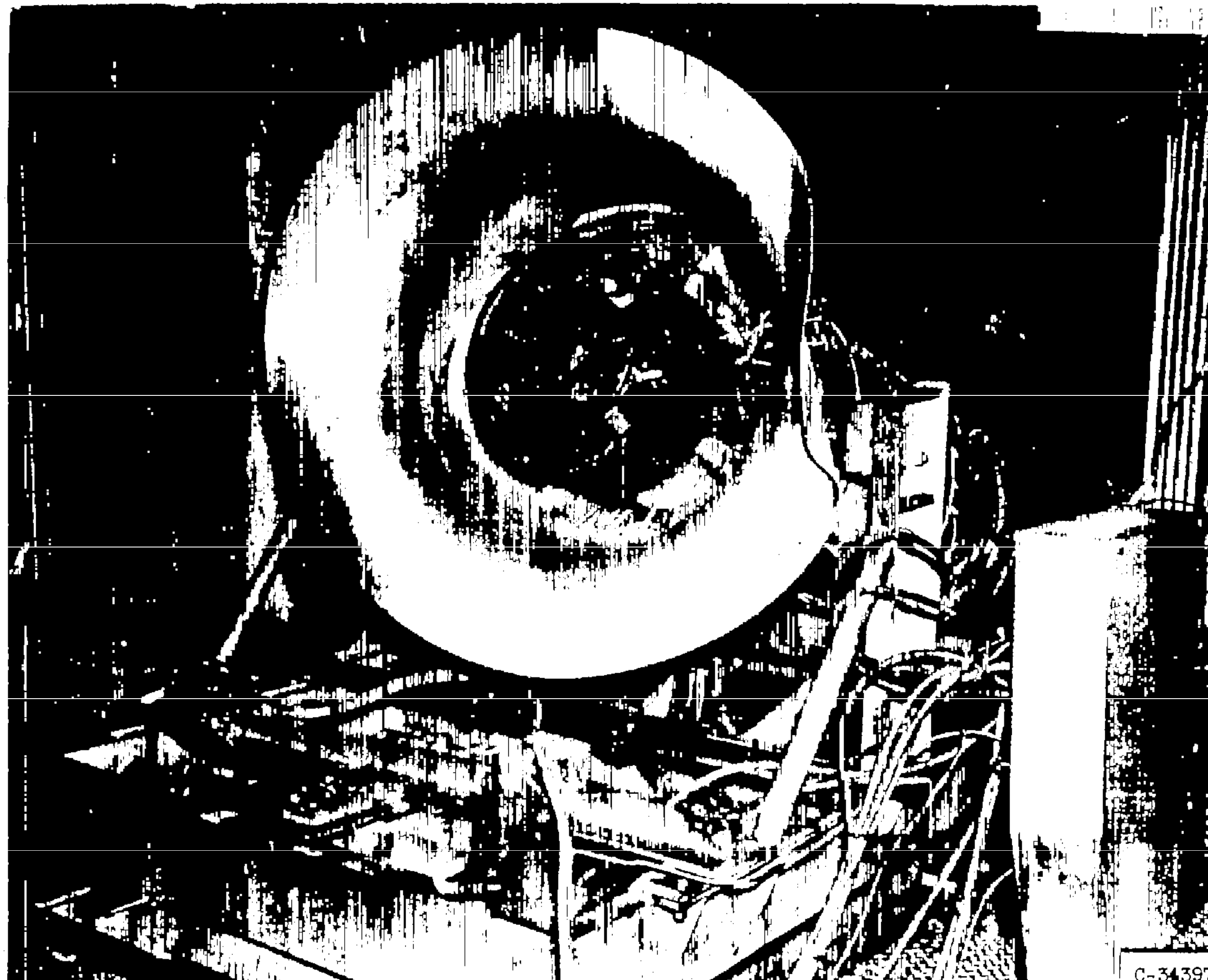

Figure 4. - Engine torn from thrust mount in test of engine employing solid-oil-stream lubrication system.

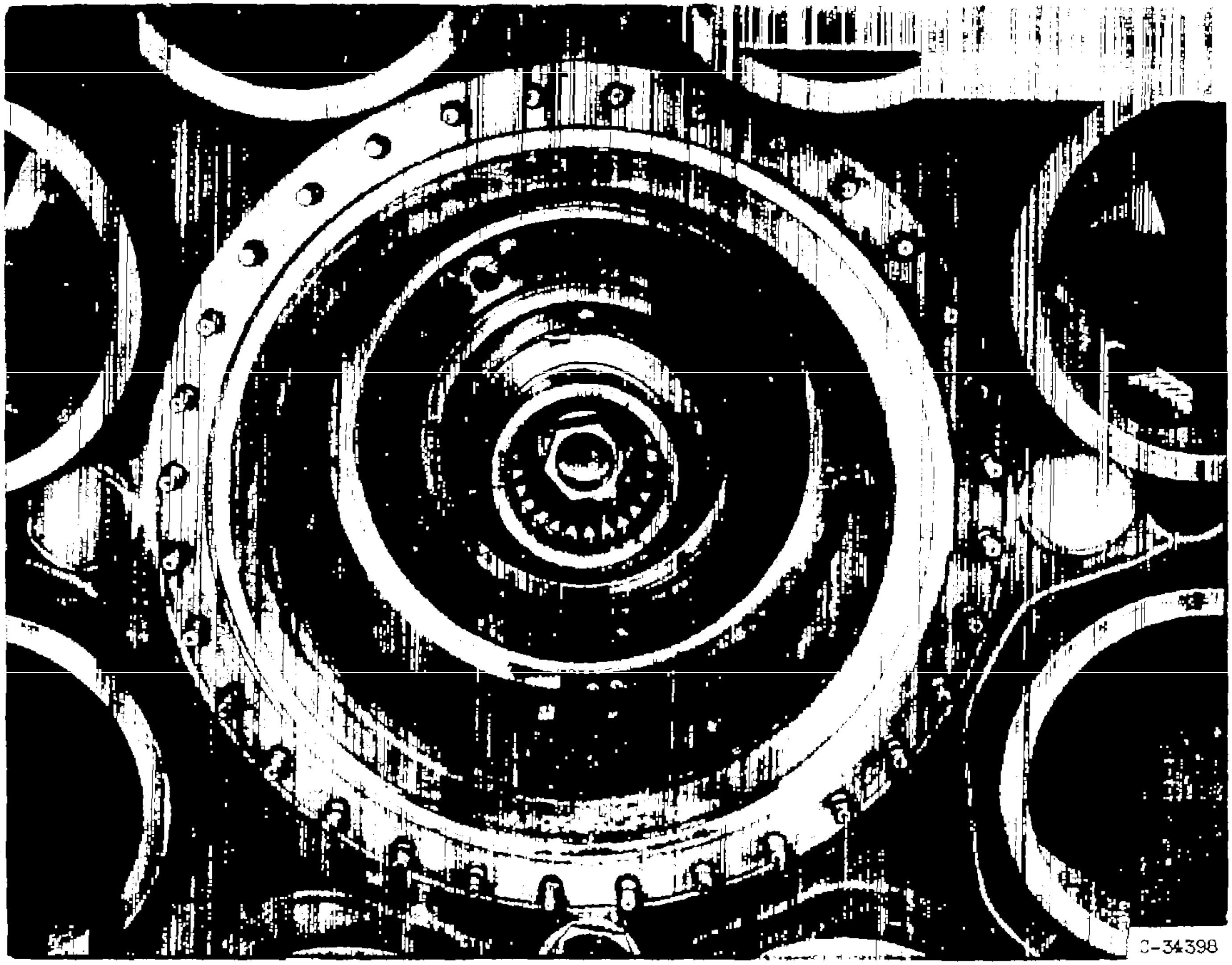

Figure 5. - Total failure of number 2 bearing of engine employing air-oil-mist lubrication system.

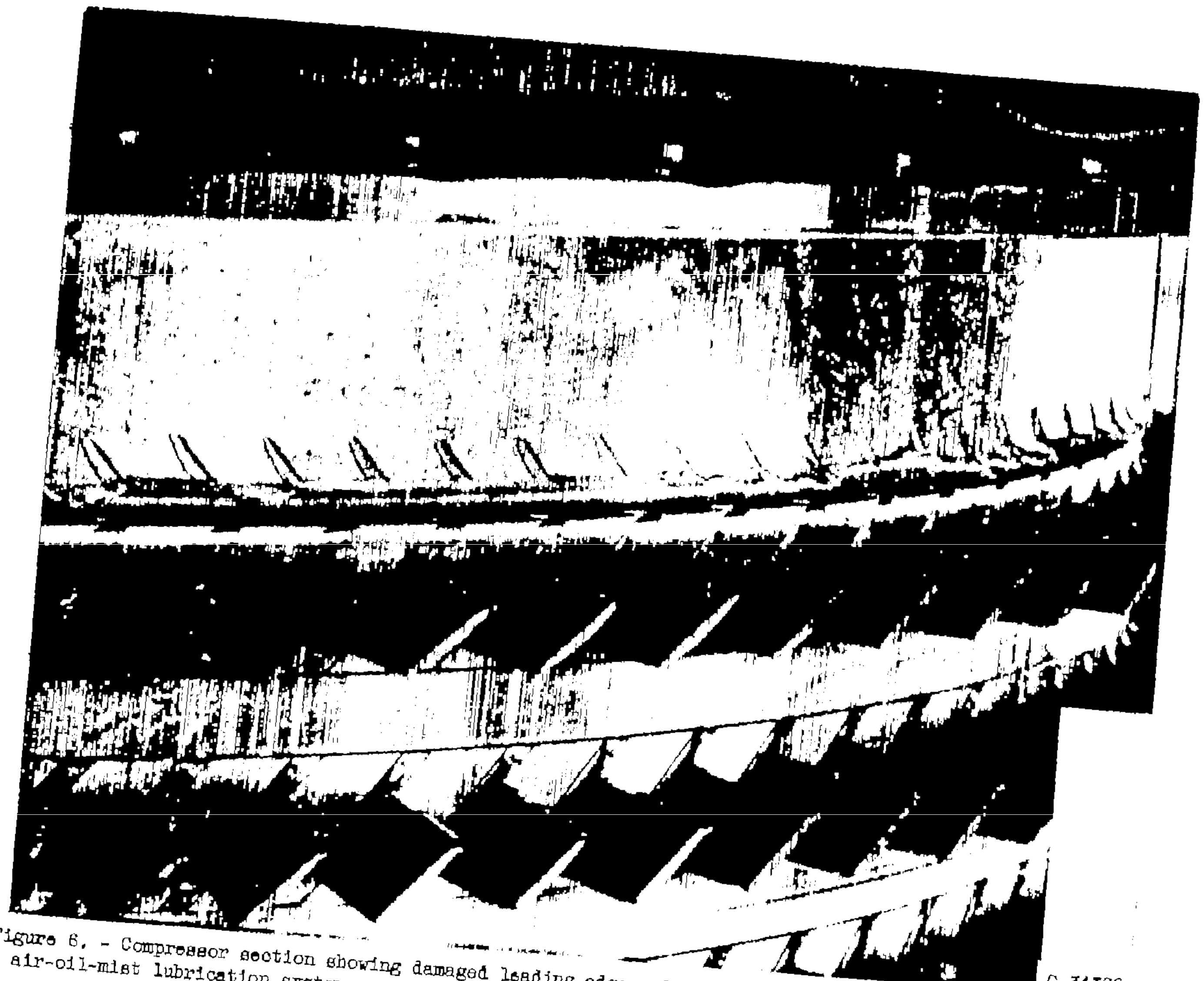

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Figure 6. - Compressor section showing damaged leading edges of compressor rotor blades of engine employing air-oil-mist lubrication system.

APPENDIX C

ROLLING-CONTACT BEARINGS AS APPLIED TO AIRCRAFT GAS TURBINES FROM THE ENGINE MANUFACTURER'S POINT OF VIEW

By Stephen Drabek
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When the designer of an aircraft turbine engine looks at his bearing problem, he is forced to give up the small, lightweight, inexpensive sleeve-type bearing for rolling-contact bearings because of (a) extreme low temperature starting, (b) low starting torque without preoiling, (c) insensitivity to oil flow interruption, (d) greater freedom in alinement tolerances, and (e) lower oil flow and cooling load. Having chosen the general type of bearing, the engine designer must then choose specific bearings with major emphasis on reliability and overall minimum weight. When in doubt, the designer will usually choose adequate reliability over a weight saving.

The present state of the art is such that rolling-contact bearings operate successfully in many thousands of jet engines at top speeds generally around 1,000,000 DN value, top temperatures about 300° F, all the radial loads that the engine can develop, and carefully minimized thrust loads.

These four areas of speed or bearing size, temperature, radial load, and thrust load are considered with respect to some of the determining characteristics, the way they affect present production units, and how the large supersonic engines of the future present some formidable problems.

Size and Linear Speed

A striving for larger diameter bearings on new, large, single rotor engines comes from a desire to increase the critical speed of the shaft to a value well above the operating range. Also, simplified designs of minimum weight can be achieved with large diameter, thin wall shafting. The limit the engine designer must first observe is the safe operating linear speed of bearings. With the present state of the bearing art he has been forced to step down the diameter of his shaft with a resultant increase in weight, complexity, and cost.

Figure 1 is shown purposely to dramatize the need for large diameter bearings. It shows an oversimplified dual rotor turbojet. Essentially it is two engines - one wrapped around the shaft of another. The low pressure unit has the long shaft. For critical speed and weight considerations, the shaft should be hollow and have a large diameter. The high pressure unit will have its hollow shaft still larger in diameter. Bearings will be placed between the two shafts. Depending on other considerations which will not be discussed, one or more bearings will probably be placed outside the larger shaft. When designers made studies of new engines with no consideration of present bearing limitations, bearing diameters exceeded 18 inches. The engine speed was not much different from speeds currently used. A quick check showed that a bearing half that size would require further development.

Thus the conflict challenges the ingenuity of the designer and results in heavier, more complicated engines on one hand and a striving for relief through development of larger bearings on the other.

Temperature

Bearing temperatures near 300° F exist in present-day production engines. This limit is a compromise of (1) the petroleum lubricating oil, (2) the common bearing steel, SAE 52100, and (3) the achievement of adequate cooling in the low pressure ratio engine operating in the vicinity of sonic speed and below.

The compressor inlet temperatures are generally under 200° F; the compressor discharge temperatures are well under 600° F. By careful design involving insulation for the bearing and oil paths and using cooling air taken from the lower compressor stages, the metal near the bearing can be brought down to about 400° F. Excess oil flow impinged against the bearing will remove the heat it generates and, in addition, lower its temperature to under 300° F.

A corresponding action takes place in the turbine area where cooling air is impinged against the turbine wheel to reduce the flow of heat to the bearing and excess oil flow removes this heat (plus the heat generated by the bearing) to keep its temperature within safe operating limits.

The problem of heat soak-back should be mentioned here. When the engine is shut down, the flow of air and oil ceases. With this cessation of cooling, the residual heat in the turbine blades, turbine wheel, and other metal parts continues to flow to the cooler parts. The turbine bearing temperature sometimes has increased more than 150° F, with the result that any oil left on the bearing is partially evaporated and the rest is oxidized. Experience has shown that a roller bearing so dried will fail after operation at up to 10 percent of rated speed in a matter of seconds. One drop of oil, however, could prevent failure.

Since with normal operation these conditions have combined to result in occasional failures, this problem was taken into the General Electric bearing laboratory for solution. Laboratory investigation showed that the bearing bronze would rub off very rapidly onto the steel race locating surface. By plating the bronze with silver the situation was greatly improved; the silver, being more compatible with steel, rubbed off much slower. This change plus changes in the oil system have combined to virtually eliminate this type of failure in practice.

The lubricant is perhaps the most important single limit to higher bearing operating temperatures. Present petroleum fluids volatilize about as fast as can be accepted in current engines. At high altitudes, this condition is aggravated by the low atmospheric pressures.

Synthetic oils are required to improve the high-temperature volatility problem without impairing low-temperature operation.

Figure 2 shows compressor temperature variation. Temperatures experienced by the 5:1 compressor ratio engines - generally well under 600° F - have been handled quite well. However, when the higher pressure ratio engine, say 15:1, is considered at sea level Mach 1 the compressor discharge temperature is about 900° F. At the cold 50,000 feet altitude this temperature reaches 1060° F at Mach 2. The lowest cooling air temperature is 300° F and is at too low a pressure for much use. At temperatures like these, new materials and some new thinking are required. Since the price of cooling quickly becomes prohibitive in supersonic aircraft, higher-temperature bearings and oils are a "must."

Radial Loads

The problem of radial load is not so important, for under normal conditions this is merely the weight of the rotor. This load is usually under 1000 pounds per bearing, which is insignificant for the size bearings used.

Gyroscopic loads due to maneuvers are very high. However, since they are within the capacity of the bearing and last for such short periods of time, the effect is one of shortened bearing life by quantities as yet unmeasured.

Thrust Loads

The problem of thrust loads is different. Here the amount of load that the bearing can take within an acceptable life period is very low. This forces the engine designer to provide balancing pistons to keep the thrust load to a minimum. These balancing pistons mean added weight, added complexity, added cost, and, perhaps most important of all, a loss of compressed air which results in shorter airplane range.

If the fact that there are two rotors in figure 1 is ignored and if the pressure on the forward area is considered as a base line, the inclined area of the compressor will have higher pressures against it, which will push the compressor backward. The compressor blades, on the other hand, are equivalent to propellers pulling the compressor forward. This results in a very high net forward force. It would be desirable to seal the compressed air between the compressor rotating surface and the stationary burners with a small diameter seal to reduce the leakage through the seal. However, this would result in a tremendous forward force due to the high pressure against this surface. Thus a large diameter seal is used near the edge to reduce this force. The result is more leakage. In some cases, the resultant forward force is still too high, so compressed air is piped to a volume in front of this front face to help balance it out. This means another seal is required at this surface and more leakage of compressed air results. Other approaches can be used, but invariably the end result is more air loss than is desired.

Also, there is a problem of balancing at all conditions of engine operation. To give some idea of the type of numbers involved, remember that these areas are of the order of magnitude of 100 to 400 square inches at the compressor inlet and 300 to 600 square inches at the compressor discharge and far greater in the blading.

Figure 3 shows that an engine with a compressor pressure ratio of 5 has a discharge pressure of 130 pounds per square inch at Mach 1 sea level and an engine with a compressor pressure ratio of 15 will develop over 400 at the same conditions. At 50,000 feet the high pressure ratio engine develops 170 pounds per square inch at Mach 2, and that pressure is climbing rapidly. The foregoing will help to gain appreciation of the fact that the engine designer is juggling forces of tens of thousands of pounds which vary with engine speed, air speed, and altitude all for the purpose of keeping the thrust load on the bearings down to a few thousand pounds.

In conclusion, the road ahead involves a large amount of development work to produce reliable bearings that will be large in diameter and which will operate at higher DN values, higher temperatures, and higher thrust loads.

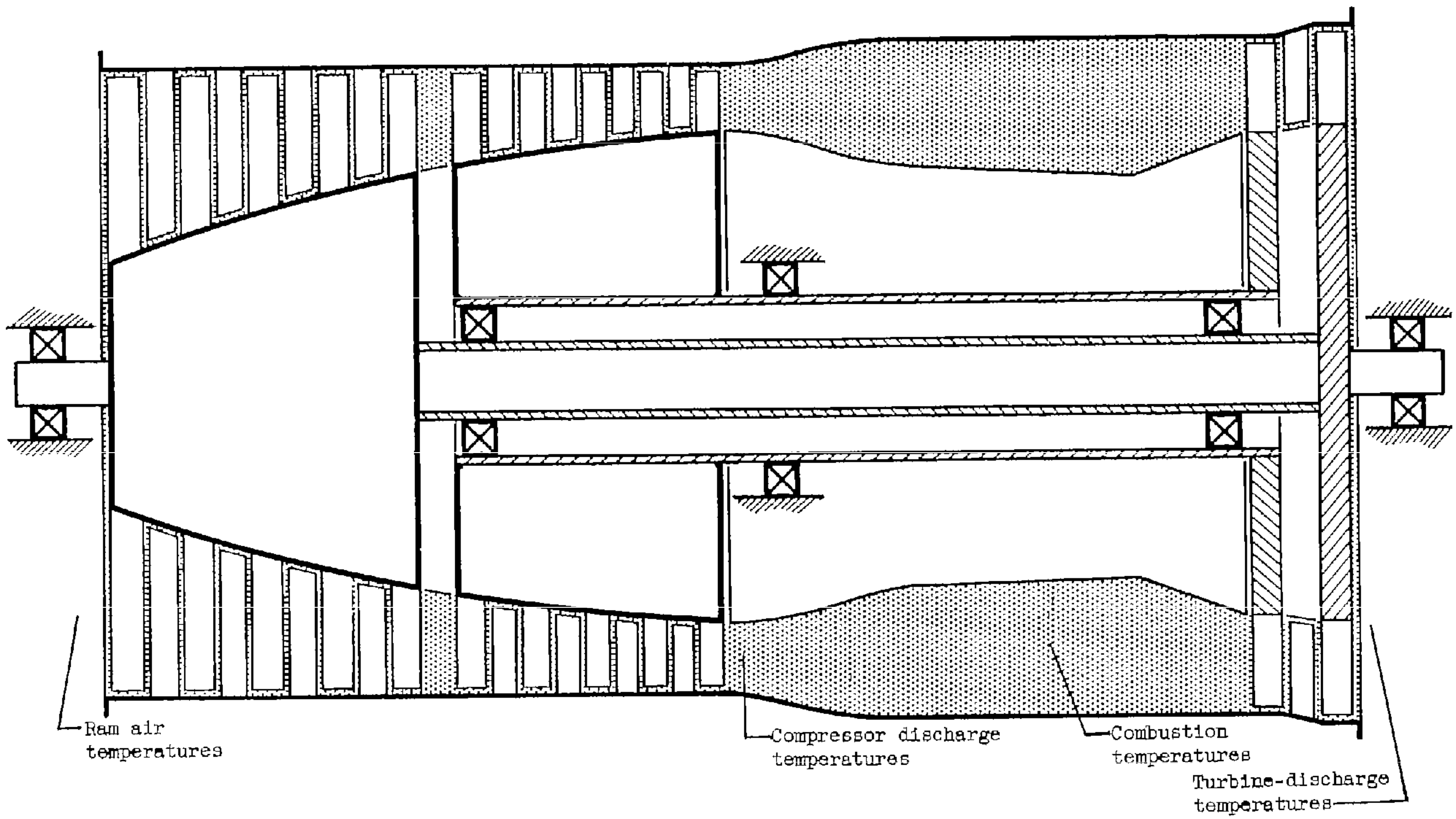

Figure 1. - Bearing location in compressor and turbine sections of dual rotor engine. CD-3604

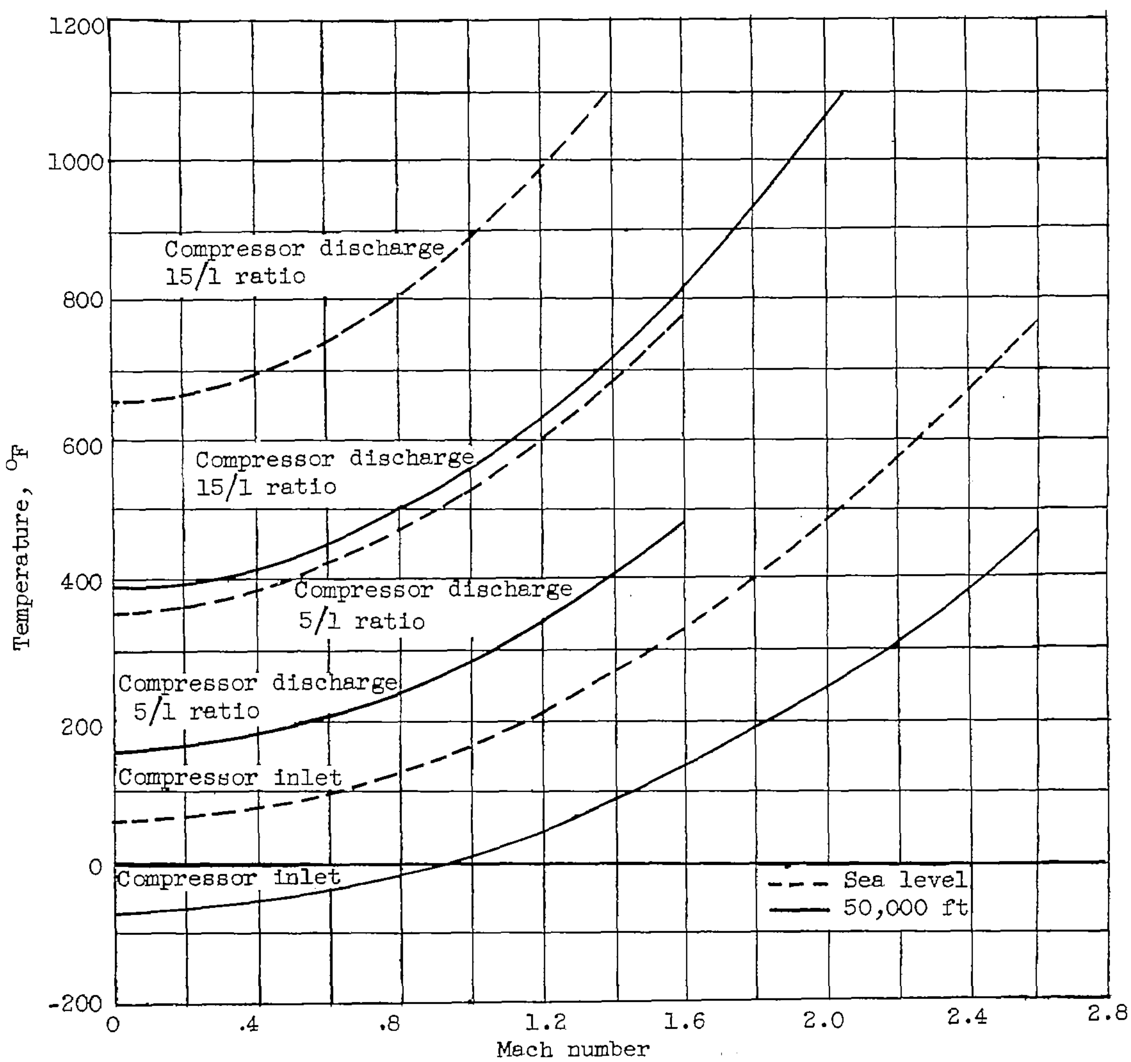

Figure 2. - Variation of compressor air temperature with Mach number.

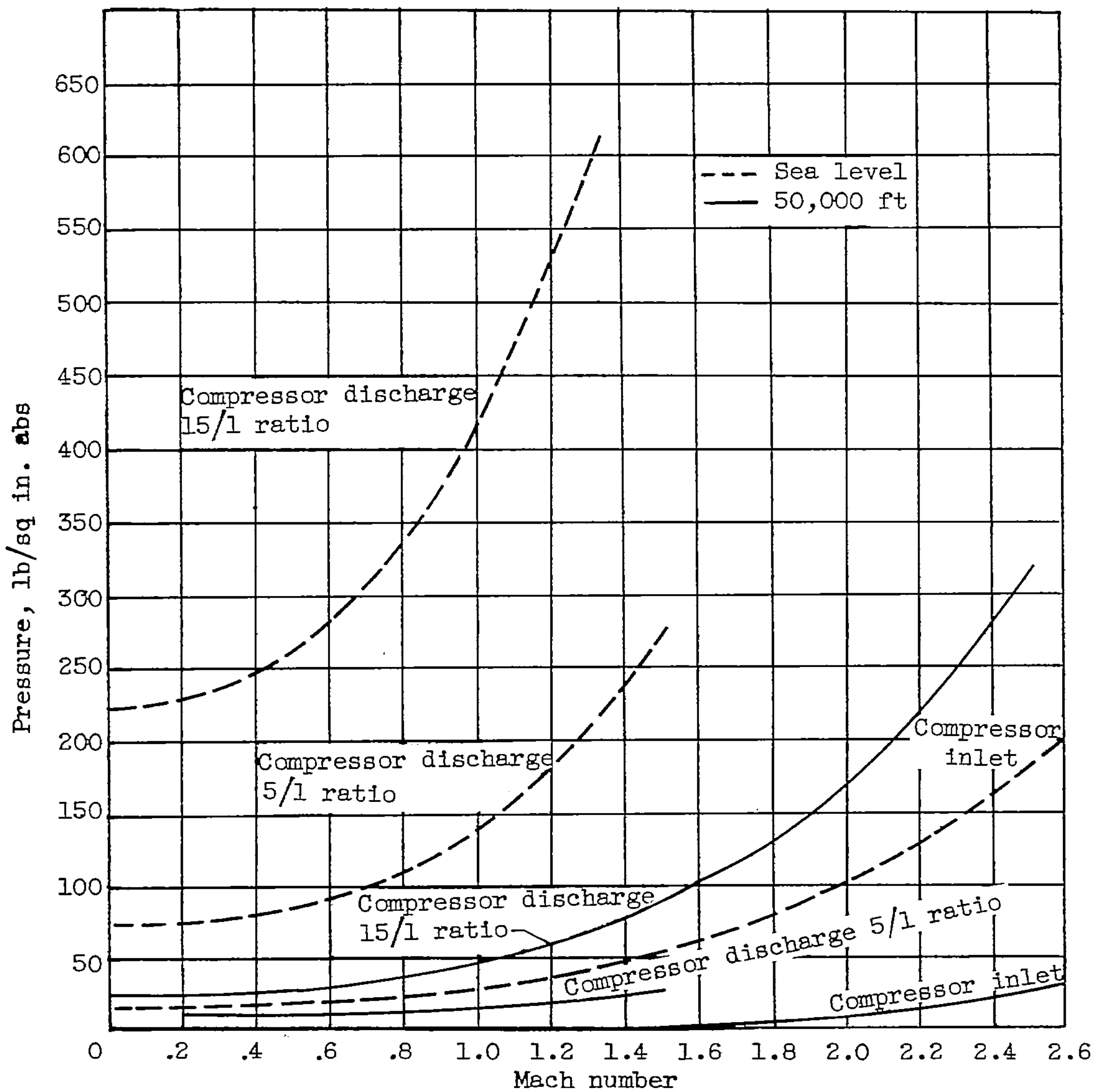

Figure 3. - Variation of compressor air pressure with Mach number.

APPENDIX D

NEW DEVELOPMENTS IN HIGH-SPEED ROLLING-CONTACT BEARINGS

By Frank W. Wellons
SKF Industries, Inc.
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With respect to bearing application and design, there is no typical aircraft turbine. However, the main shaft carrying the turbine and compressor is supported on two or more bearings and there is a multitude of essentially standard catalog ball and roller bearings in the gear case and its related accessory drives. The speeds, loads, and temperatures are generally most severe for the main shaft applications, so that by comparison, very little attention has been paid to the more conventional bearings in the gear case, although these applications represent by far the largest number of bearings in the engine. For convenience, we will limit our consideration of bearing problems with high performance engines to the main shaft applications, although it must be recognized that the trends toward higher speeds, loads, and temperatures have an increasing effect on all other bearings in the engine.

Bearing speeds are generally measured in terms of the DN value for a given application where D is the bearing bore in millimeters and N, the shaft speed in rpm. The initial bearing applications in jet engines were at DN values of approximately 800,000; but as newer engines have developed, there has been a fairly constant increase in both shaft speed and bearing bore, so that today DN values approaching 1.5×10^6 are a reality. This raise in the DN limit has been accomplished by numerous refinements in bearing design, materials, lubricating provisions, and so forth, and it appears that the trend toward even higher DN values will continue, although possibly at a lesser rate. In any case, there is no indication of a sudden jump in bearing speeds and there is no reason to expect that engine development will suddenly be retarded by inability of the main shaft bearings to operate at optimum design speeds.

A second consideration for future engine programs is the effect of synthetic lubricants on the load carrying ability of antifriction bearing components. Ball and roller bearings are selected for a given life expectancy under the anticipated load and speed. An effectively protected and well lubricated ball or roller bearing will operate indefinitely until repeated stresses initiate subsurface cracks which develop into spalling of the loaded surfaces. Endurance tests conducted under standardized conditions in the SKF Laboratory have given an indication that the fatigue life of the bearing parts can be affected by the type of lubricant (see fig. 1). Groups of bearings were tested with various freak and unorthodox lubricants such as cutting compounds, gasoline, and so on, and the average life was much less than for similar test groups of the same size bearings operating under identical conditions but lubricated with conventional oils and greases. Premature fatigue failures of this nature have never been apparent with current engines, probably because the bearing lubricant must also lubricate the gear train; but in screening possible fluids for high-temperature operation, consideration should be given to the effect on the load carrying ability of the ball and roller bearings.

The maximum outer race temperature for satisfactory operation of bearings made from conventional materials such as SAE 52100 is generally considered to be 350° F. To take full advantage of synthetic lubricants and to avoid bearing damage from excessive soak-back temperatures after engine shutdown, it will be necessary to consider new steels for the main bearings of future turbine power plants. In selecting the most suitable analysis for these high-speed, high-temperature bearings, there are several characteristics that must be carefully weighed:

1. An analysis must be chosen which can be heat-treated to obtain an acceptable hardness and yet maintain dimensional stability at maximum operating temperature.

2. For ultimate load carrying capacity, a minimum hardness of at least Rockwell C-58 must be maintained at the maximum operating temperature, and the hardness must be essentially unaffected by extended service at this elevated temperature.

3. A suitable high-temperature steel should require a minimum of critical alloying elements with particular emphasis on tungsten.

4. The analysis should be available from several competitive sources and covered by a recognized industry specification.

5. Although it is most likely that any bearing suitable for high-temperature operation will be more difficult to manufacture than one made from 52100, consideration should be given to heat-treatment and grinding characteristics.

In England 18-4-1 tool steel has been used for high-temperature applications, but in the United States a strong effort has been made to use analyses with the necessary high-temperature properties but having a lower critical alloy content. A great many high-temperature bearings have been made from the M2 analysis with 6 percent tungsten, but post-Korea thinking has leaned toward the M10 steel which requires no tungsten. Oddly enough, the mills which furnish M10 will show up to 1 percent tungsten as an impurity because of the type of scrap they are obliged to use. Therefore, a third analysis, M1, with $1\frac{1}{2}$ percent tungsten has recently been considered both because of its more favorable manufacturing characteristics and because it is preferred over M10 by the steel makers. The compositions of these various bearing steels are as follows:

	52100	M2	M10	M1
C	0.95 - 1.10	0.75 - 0.85	0.83 - 0.93	0.75 - 0.85
Mn	.25 - .45	.20 - .40	.10 - .35	.15 - .40
Si	.20 - .35	.20 - .40	.20 - .40	.15 - .40
Cr	1.30 - 1.60	3.75 - 4.50	3.75 - 4.50	3.50 - 4.25
P (max.)	.025	.030	.030	.030
S (max.)	.025	.030	.030	.030
Ni (max.)	.35	-----------	-----------	-----------
Cu (max.)	.25	-----------	-----------	-----------
Mo	.08 (max.)	4.75 - 5.25	7.75 - 8.50	8.25 - 9.50
W	-----------	5.50 - 6.50	1.00 (max.)	1.30 - 1.80
V	-----------	1.75 - 2.25	1.80 - 2.15	.90 - 1.25
Temper, °F	455 - 465	1050 - 1100	1050 - 1075	1025 - 1075
Hardness, Rockwell C	58 - 61	62 - 65	62 - 65	62 - 65

Dimensional studies have been made with M1, M2, and M10 rings by heating at various temperatures up to 900° F for 2 hours and then cooling to room temperature overnight. For all three analyses there was no diameter change through 700° F and a negligible change up to 900° F.

Hot-hardness tests have been made for a great number of steels including the normal bearing steels, intermediate high-temperature steels, and the M series tool steels. Referring to figure 2, it is noted that 52100 steel which has been given stabilizing heat-treatment to minimize growth at elevated temperature will maintain a fairly constant hardness up to 400° F and then fall off quite rapidly. The hardness for these tests was obtained by initially bringing a small specimen up to the desired temperature, rapidly transferring the piece to a holding furnace mounted on the hardness tester, and then taking conventional hardness readings with the specimen at temperature. The hot-hardness for M series tool steels which have been dimensionally stabilized by a double draw at approximately 1050° F falls off fairly uniformly with temperature to a hardness of Rockwell C-57 at 900° F. Because of the cost and manufacturing penalties attached to the M series steels, various intermediate analyses have been evaluated for applications where the operating temperature would only slightly exceed the limitations for 52100. These intermediate analyses, having more conservative alloy content, would be much easier to fabricate and yet would permit a currently ample 200° F increase in operating temperatures, as demonstrated by the hot-hardness curves on figure 2.

For lightly loaded main roller bearings where capacity is unimportant, a limited increase in operating temperature can be made if a lower-than-normal hardness can be tolerated. For an application at 500° F, rings and rollers for an experimental bearing have been dimensionally stabilized by a 2-hour draw at 600° F which resulted in a hardness of Rockwell C-55.

Although the decreased capacity is of no great concern, the soft rings and rollers will be susceptible to nicks and dents from handling abuse which might cause premature failure.

Results of grindability tests using 52100 as a reference are shown on figure 3. M1, the best of the high-temperature steels (from a grindability standpoint), required 80 percent longer time for the same stock removal as a comparable operation with a similar ring of 52100. For simplicity, the several intermediate steels covered by this investigation have not been listed on the chart, but generally speaking, these analyses have not shown much better grindability than M1. For a more comprehensive production comparison of high-temperature steel bearings, comparative times for making a complete roller bearing are shown in figure 3. The experimental analysis at the extreme right of figure 3 was developed for ideal hot-hardness and low alloy content and is noteworthy as the most difficult steel to grind that has yet been considered for bearing usage.

For some time to come, it is likely that a variety of steels, that is, either conventional steels with modified heat-treatments or suitable analyses for the operating conditions, will be used in aircraft turbines. However, as the trend toward higher temperatures continues and as new lubricants become available, high alloy tool steels will be obligatory for reliable performance. With this inevitable goal in mind, it would appear to be good economy to concentrate attention on the manufacturing problems and performance characteristics associated with those analyses which will be of long range benefit.

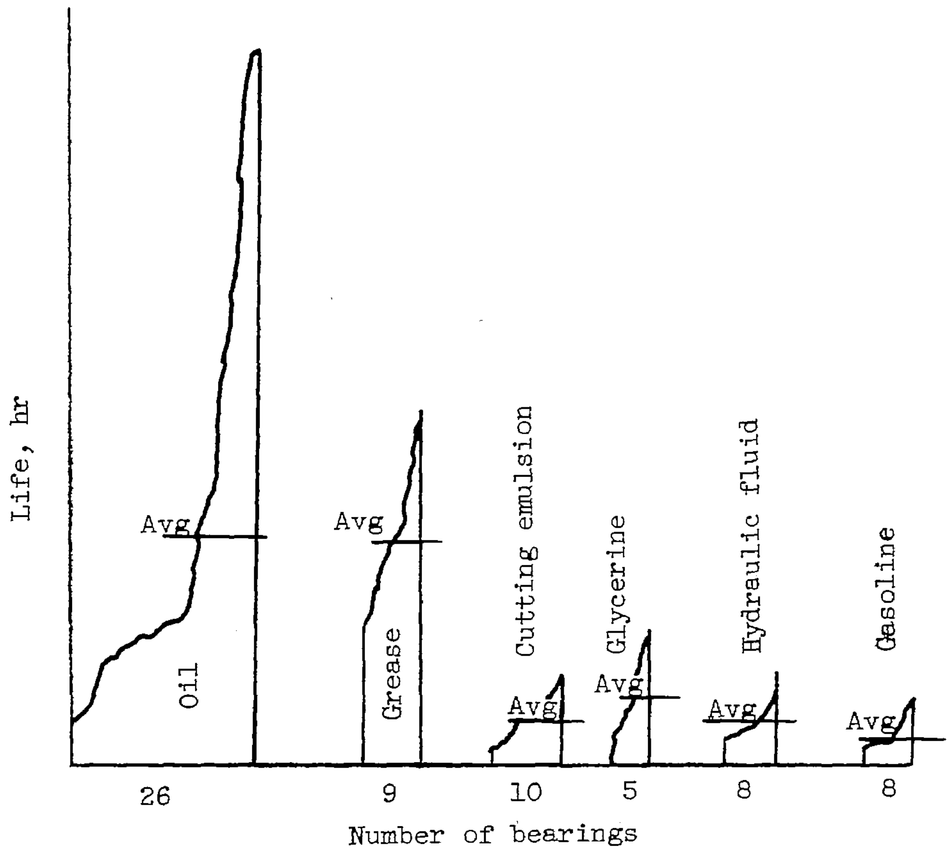

Figure 1. - Endurance tests with various lubricants.

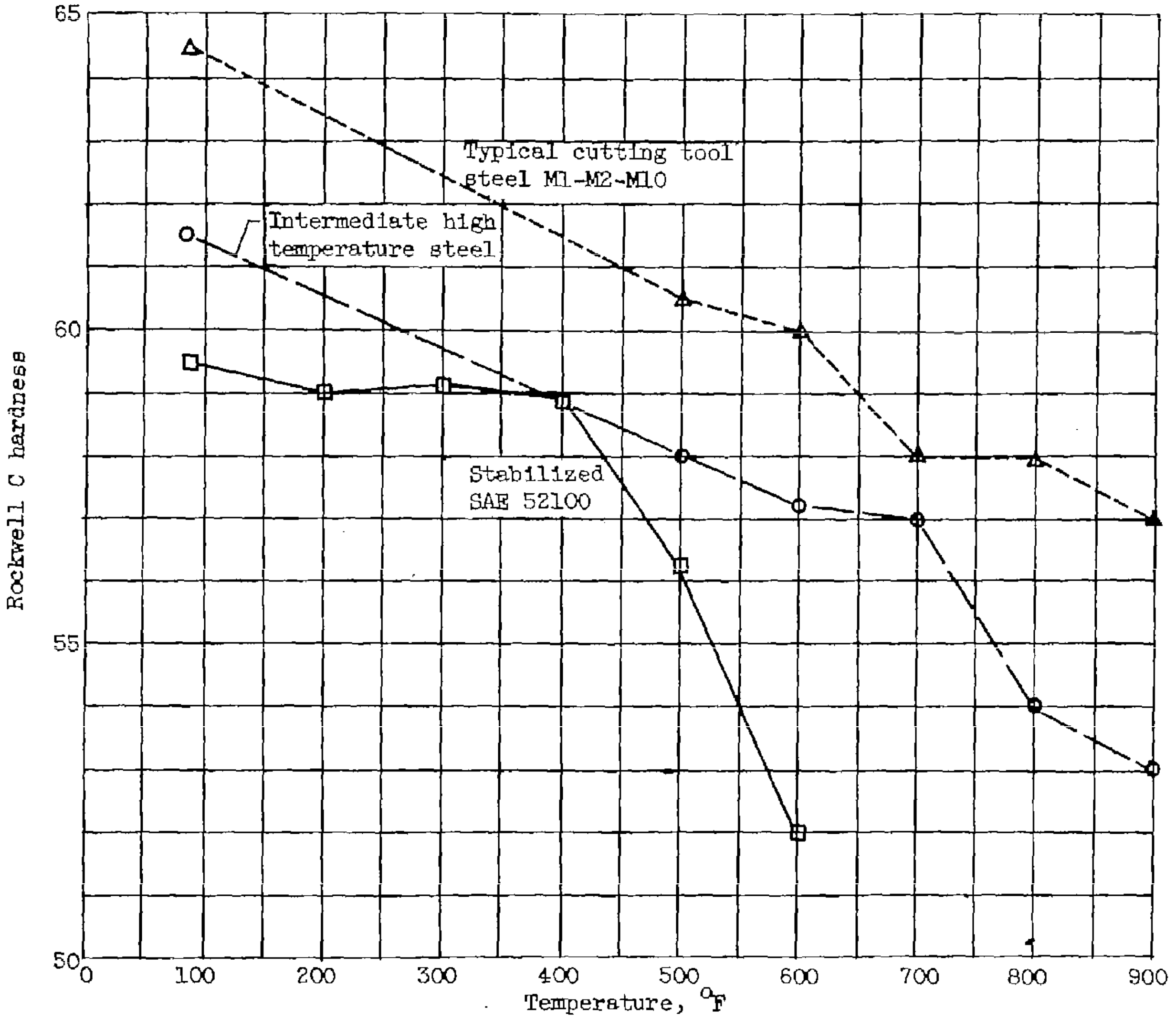

Figure 2. - Hot-hardness for bearing steels at elevated temperature.

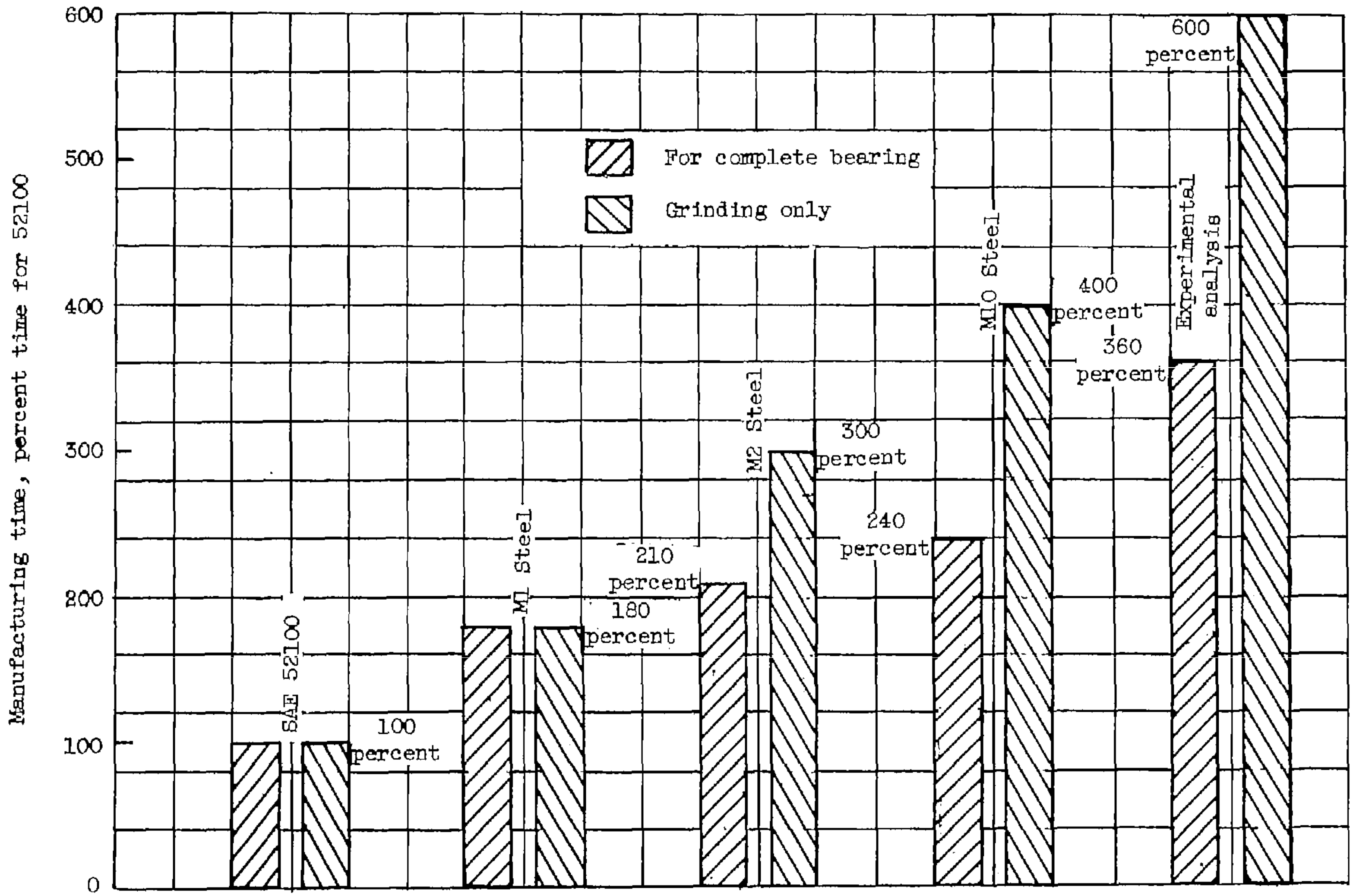

Figure 3. - Relative manufacturing times for a typical turbine roller bearing when made from different steels.

APPENDIX E

BASIC FRICTION AND WEAR STUDIES OF ROLLING-CONTACT-BEARING CAGE MATERIALS

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INTRODUCTION

One of the principal sources of failure in rolling-contact bearings has been the cage (separator or retainer). Research and service experience has shown that these failures are lubrication failures at the cage locating surfaces caused by inadequacies in the lubrication resulting from several factors. The high-temperature "soak-back" (after shutdown) of the turbine bearing causes vaporization of the residual lubricant and leaves the bearing surfaces relatively clean and dry. In consequence, during subsequent starts of the engine these surfaces (some of which are in pure sliding) are operating under conditions of dry friction. A second and very important factor which results in lack of lubrication is that caused by oil interruption, that is, interruption to the flow of lubricant by loss of the lubricating system.

One of the methods of approach to this type of problem is to use materials for cages which are inherently "nonwelding," even when operating in the extreme boundary or dry conditions. An investigation was therefore made of some possible cage materials. This investigation was for the purpose of studying the friction and wear properties of these materials when operated under pure sliding conditions, both dry and lubricated.

The results contained in this discussion are based on the data of reference 1.

MATERIALS

The reasons for selection of the various materials were based on both present and anticipated requirements for cages. The materials selected, in general, had mechanical properties that compare favorably with those of materials in present use. Since the actual strength requirements for a cage material are unknown, a strength equal to that for commonly used materials was chosen as a standard.

One of the extremely important physical properties of possible cage materials is the thermal expansion coefficient; the thermal expansion coefficient should be as close to that of the race materials as possible in order that differential expansion difficulties may be minimized. Another important factor, particularly for high-temperature bearings, is that of corrosion or oxidation resistance.

RESULTS

The friction apparatus for these experiments consisted essentially of a rotating disk specimen and a rider specimen, with the two specimens in pure sliding only. This apparatus is shown in figure 1. All wear runs were made without radial traverse so that the rider specimen was sliding in the same track on the disk specimen. In all cases the disk specimens were of SAE 52100 steel and the rider specimens were made from the different materials investigated. The rider specimen was elastically restrained through a dynamometer ring assembly, and friction force was measured by use of strain gages on the dynamometer ring. Wear was recorded by measuring the wear spot diameter on the rider specimen at regular intervals.

The wear data for the various materials are shown in figure 2 (50g load) and figure 3 (269g load). Comparison of these two figures shows the extremely important effect of load, since some of the materials which appeared fairly promising at the light loads show extremely high wear rates at high loads, for example, monel. These data may help to explain why monel has shown erratic behavior in service. It is entirely possible that, under certain conditions, the monel cages have been operating under reasonably low surface loads and have therefore operated successfully. In other instances, the surface loads may have been high and the monel may have failed as a result.

Examination of the wear tracks on the 52100 disk specimen revealed that cast irons were the only materials that did not have appreciable metal transfer to the disk surface. The cast irons formed black films which may have been graphitic carbon. No severe scoring of the steel surfaces was observed in the dry runs with any of the materials.

During the dry wear runs there were indications that films were forming on the surfaces of some of the specimens. The films were not stable, however, and showed continual buildup and breakdown. The friction and wear properties of the materials were appreciably affected by the presence of the film; in general, good friction and wear properties were obtained when the film was present. It is believed that the film from bronze was formed by lead from within the structure of the bronze and that the films on nichrome, monel, and beryllium copper were oxide films.

Figure 4 shows the wear areas of rider specimens after the dry wear runs at the 269 gram load. Visual observation of the wear areas showed partial film formation on the bronze, beryllium copper, and nichrome and also showed evidences of plastic deformation of the monel as well as of the very good surface appearance of the cast irons.

In general, the dry friction coefficients of the various materials against 52100 steel were relatively high except for bronze which had a low friction coefficient at low sliding velocities but showed an upward trend (fig. 5).

Under lubricated conditions there was very little difference in friction coefficient for all materials except beryllium copper, which showed an extremely high friction coefficient (fig. 6) at high loads.

SUMMARY

Under the conditions of this investigation the results indicated:

1. The ability of the materials to form surface films that prevent welding is a most important factor in both dry friction and boundary lubrication. These surface films were probably supplied from within the structure of the cast irons by graphitic carbon and of the bronze by lead. Monel, nichrome, and beryllium copper formed films believed to be oxides under both dry and lubricated conditions.

2. On the basis of wear and resistance to welding only, the cast irons showed appreciable promise. These materials are promising also because they have thermal expansion coefficents nearly equal to that for 52100 steel and those for some tool steels.

REFERENCE

1. Johnson, Robert L., Swikert, Max A., and Bisson, Edmond E.: Investigation of Wear and Friction Properties Under Sliding Conditions of Some Materials Suitable for Cages of Rolling-Contact Bearings. NACA Rep. 1062, 1952. (Supersedes NACA TN 2384.)

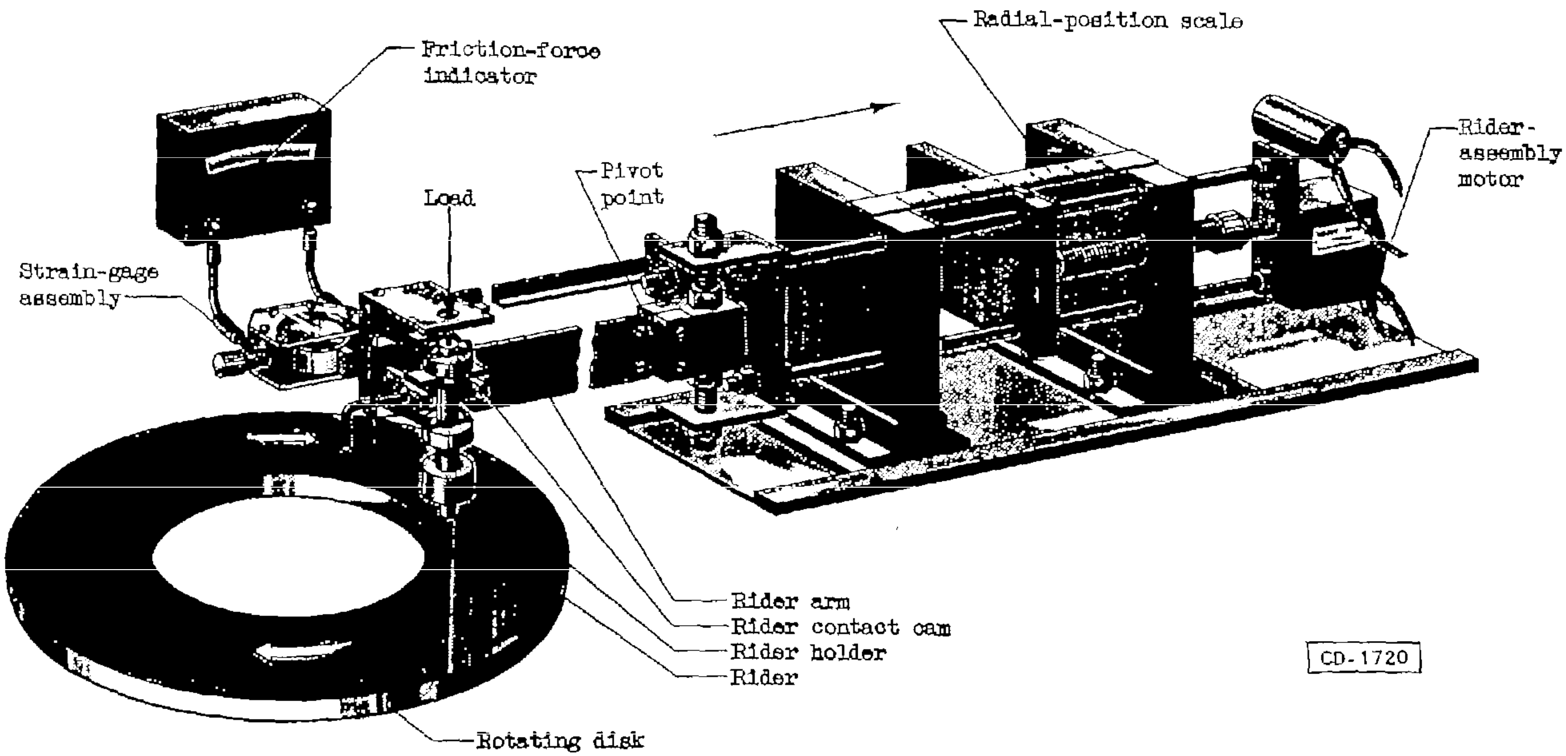

Figure 1. - Schematic diagram of apparatus.

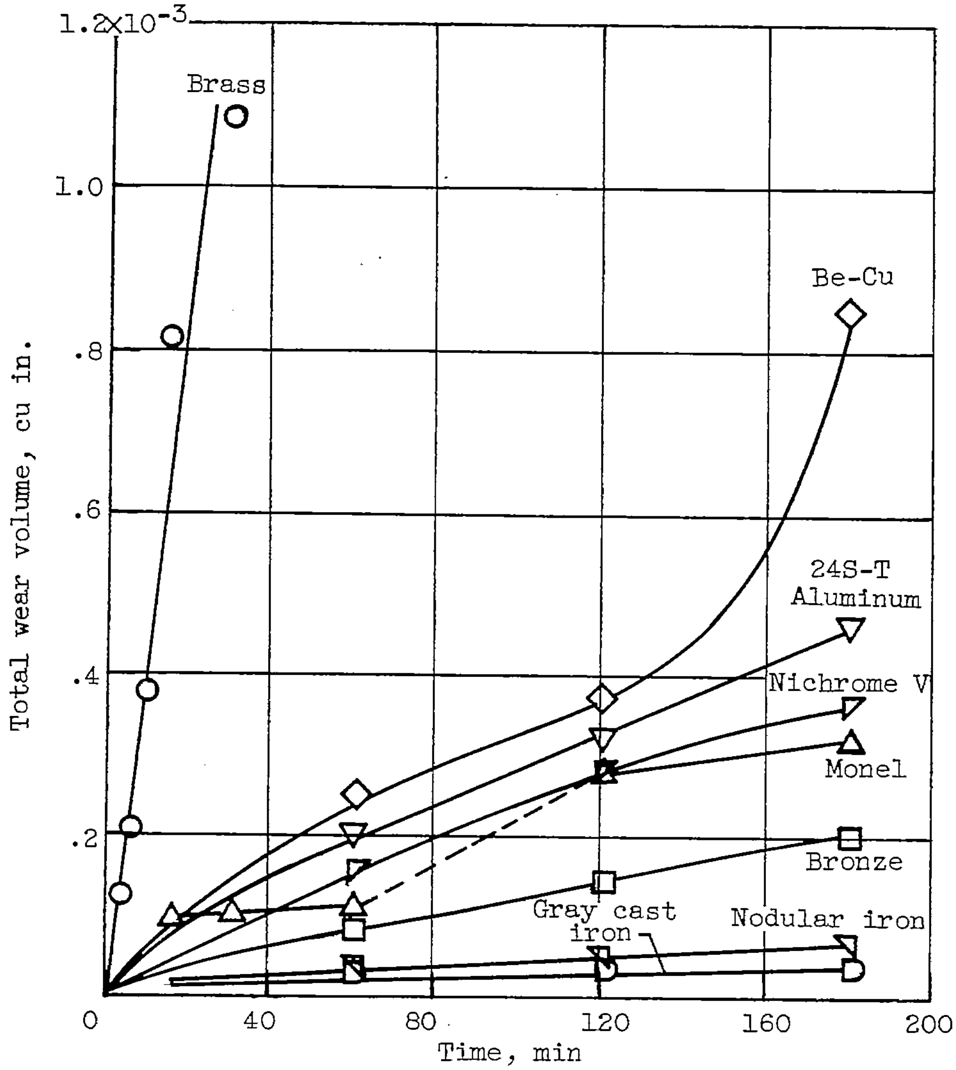

Figure 2. - Wear curves. Various materials against 52100 steel; load, 50 grams; sliding velocity, 5000 feet per minute.

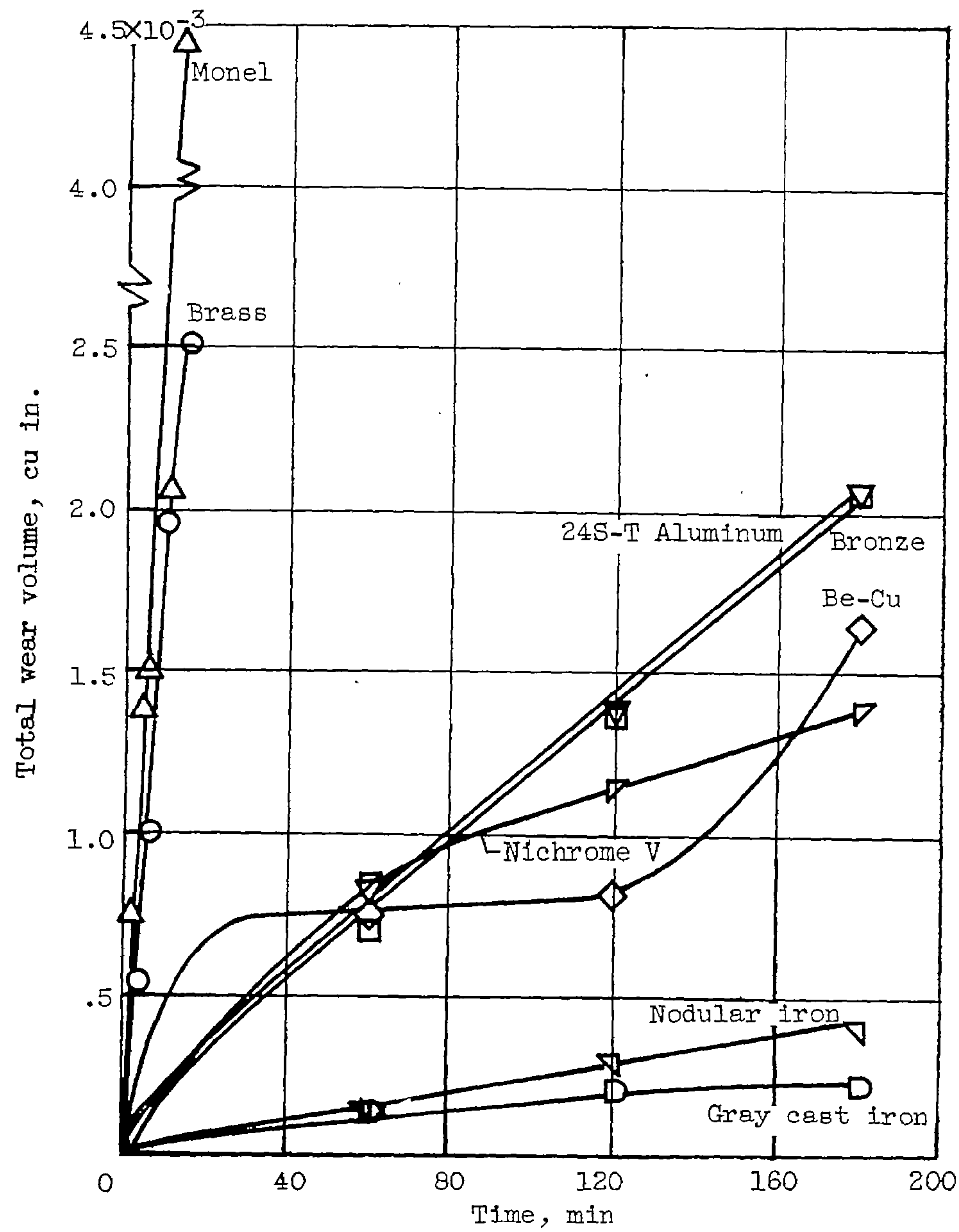

Figure 3. - Wear curves. Various materials against 52100 steel; load, 269 grams; sliding velocity, 5000 feet per minute.

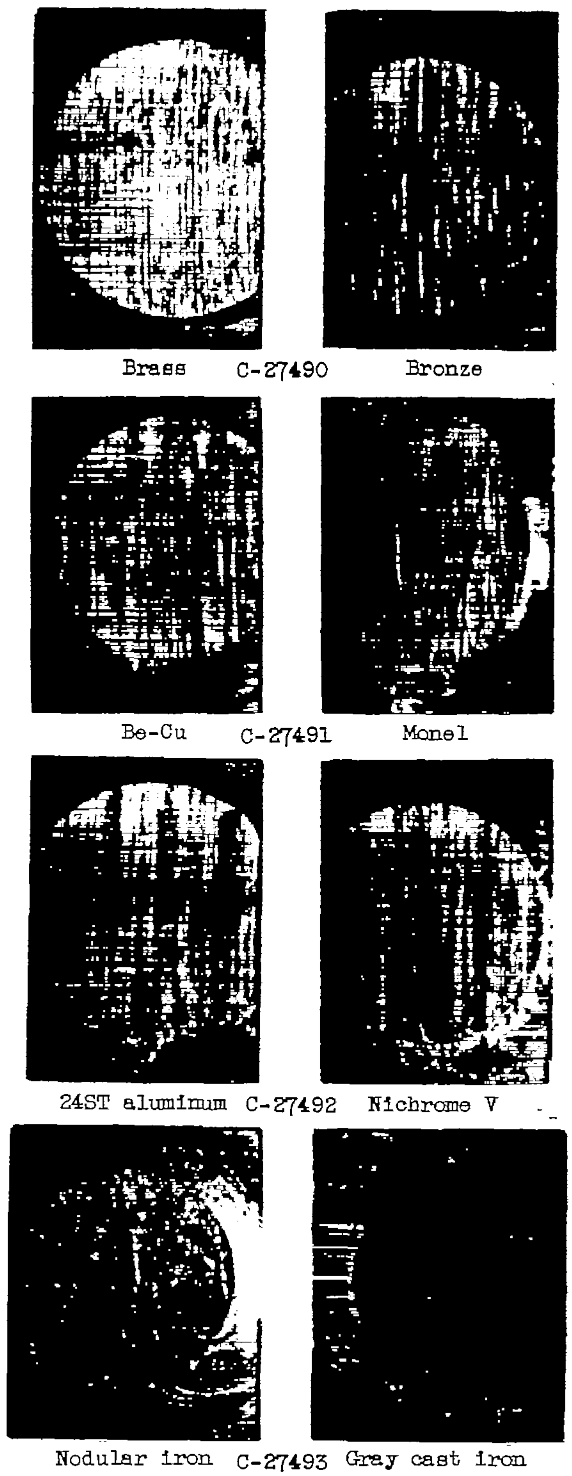

Figure 4. - Wear areas of rider specimens after dry runs.

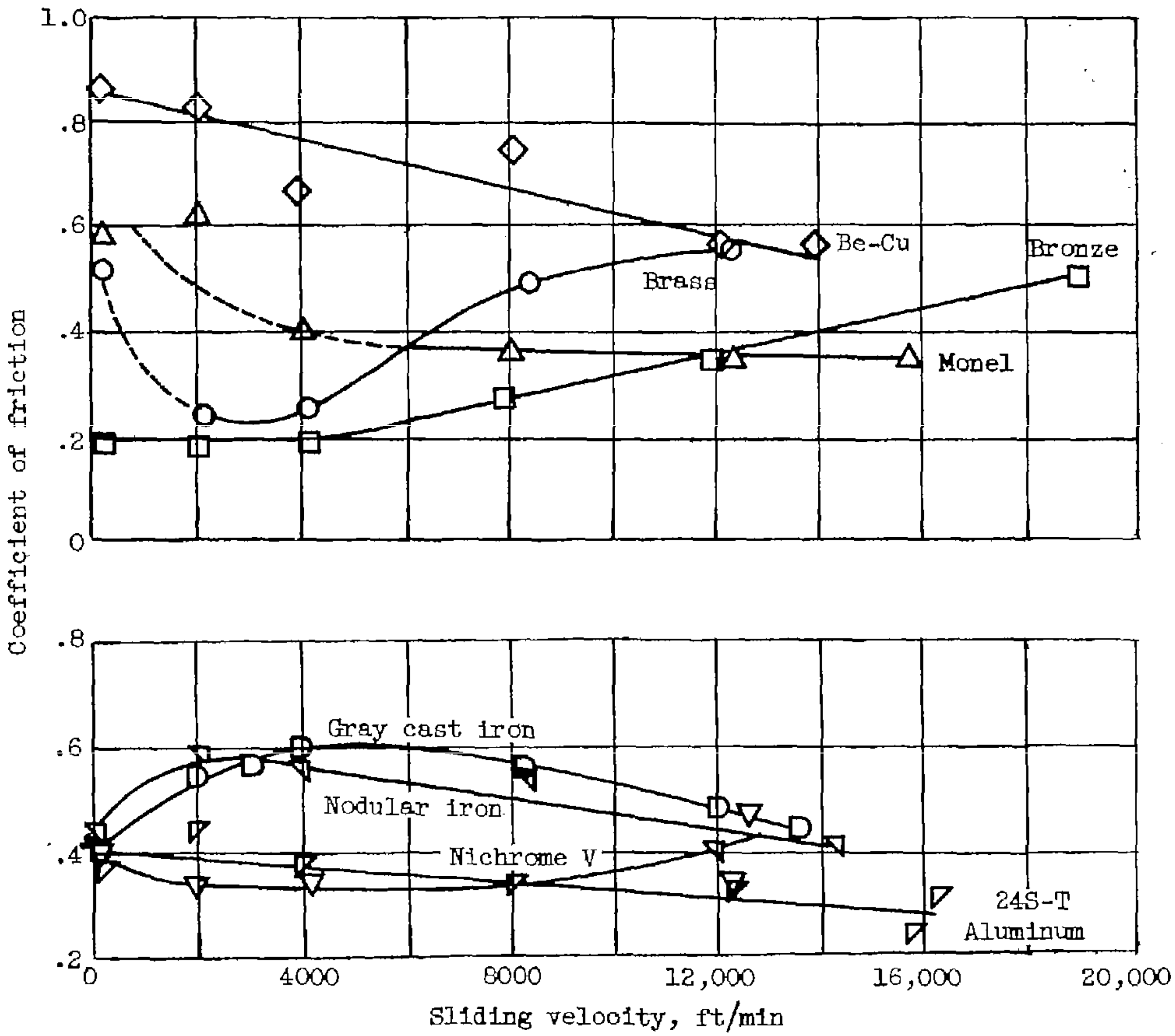

Figure 5. - Friction of unlubricated specimens. Load, 100 grams.

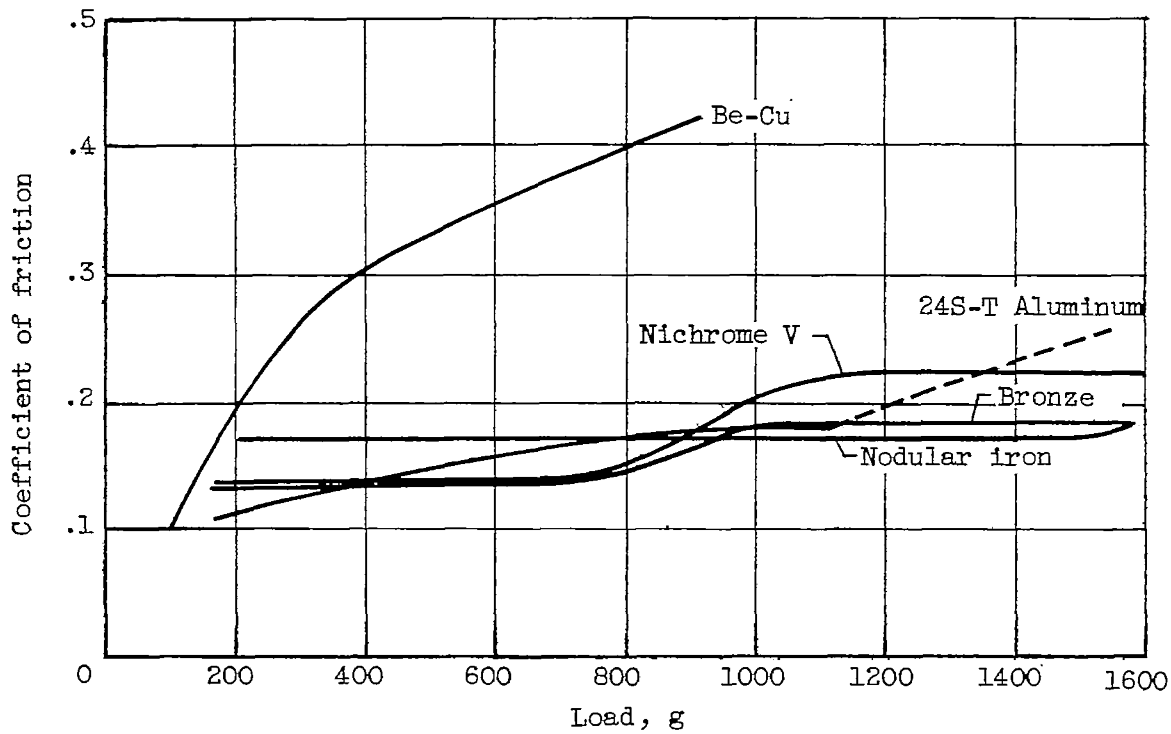

Figure 6. - Friction of lubricated specimens. Sliding velocity, 5000 feet per minute.

APPENDIX F

PRESENT STATUS OF RESEARCH KNOWLEDGE IN THE FIELD OF HIGH-SPEED ROLLING-CONTACT BEARINGS

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The status of research knowledge in the field of high-speed rolling-contact bearings is gradually increasing. However, the demands of new aircraft engines are far in advance of present bearing research knowledge. Most of today's engines are being designed from extrapolated bearing knowledge. In many cases the bearing is a limiting feature in the design, as designers admit they cannot venture too far into the unknown.

Among the most critical bearings in the aircraft engine are those that locate the main rotor in both the radial and axial directions. Ultrahigh surface speeds, increasing temperatures, and increasing thrust loads are the conditions which are making the problem increasingly critical. Limitations of the lubricant are apparent and will not be discussed here. The rolling-contact bearing limitations are due both to inherent characteristics of this bearing type and to limitations of the available materials.

Figure 1 gives a rough idea of the state of knowledge regarding high-speed, high-temperature, radial-load bearings. The rolling-contact bearings best suited for the subject application are cylindrical roller bearings and deep-groove ball bearings. The abscissa shows the bearing speed in both DN value and feet per second. In each case the values are based on the bearing bore and shaft speed. The ordinate gives the operating temperature in degrees Fahrenheit. A decade ago, a bearing operating at 100 feet per second was considered to be a high-speed bearing. The future objective requirements are indicated by the dashed lines. The known area is shown in the lower left-hand corner. Not everything is known about bearings even in this area. Enough is known, however, to allow for safe operation of bearings in engines within the operating conditions covered by the area.

The maximum known engine and research data are indicated. These data do not constitute very much tangible in the way of reliability, as only a few hours of operating time have been accumulated at most of these points. These points mean only that bearings have operated

[1]Presently a consulting engineer.

at these extended conditions. It is seen that bearings have operated either at DN values as high as 2.4×10^6 or at temperatures just above 500° F.

It is also seen that bearings have operated at temperatures to 1000° F at low speeds. As may be seen, the trend is toward the upper right-hand corner, that is, to the highest speed and highest temperature; it is obvious there is a long way to go.

Figure 2 illustrates in a broad sense the state of affairs regarding the high-speed, high-thrust-load rolling-contact bearings. The angular-contact ball bearing is generally considered the best for this application. The abscissa is again given in both DN value and feet per second. The ordinate is given as thrust load in pounds. The future objective requirements are again indicated by the dashed lines. The known area is indicated on the left portion of the figure. The maximum known engine and research data are also given; here again the data show only that a bearing has been run under the indicated conditions of operation. The trend is indicated toward the upper right-hand corner of the figure, and again it is seen that there is a long, long way to go. The data show that bearings have operated at DN values as high as 3.1×10^6 under very light thrust loads and as high as 1.4×10^6 under thrust loads of 18,000 pounds.

Figure 3 is a bar graph illustrating the rolling-contact bearing material situation with respect to the future objective temperature requirement of 1000° F (indicated by the dashed line on the right). The abscissa is given as the operating temperature in $^\circ$F; the materials considered are for races and rolling elements in one case and for cages in the other. It is seen that materials are available for races and rolling elements to temperatures in the neighborhood of 450° F without use of strategic materials. Above 450° F some penalty must be paid in strategic materials. There are some engine operating data in the region below 500° F. However, in the region above 500° F, there are practically no research or engine data. The graph above 500° F is based purely on the physical properties of materials. In the case of materials for cages, operation in engines has reached about 500° F. The search for cage materials is continuing. Several materials look promising above 550° F, although no research data or engine data have been obtained to verify this.

Up to now life and reliability have not been discussed to any extent. These factors are of utmost importance in aircraft. There are two vastly different factors affecting life and reliability for the subject application: one is wear and the other is fatigue. Even with additives in the oil, roller bearings wear beyond acceptable limits under the operating conditions of some of today's experimental engines.

External radial loads are not expected to be a cause of fatigue in ball or roller bearings. However, at the ultrahigh speeds, the rolling element centrifugal load is so great as to limit the life of the bearing by fatigue at zero external load. A rough calculation has been made as to what this limiting DN value is for 1000-hour life based on 90 percent survival. The calculated answer is about 2×10^6 DN or 350 feet per second for a 45-millimeter bore ball bearing. This figure is based on the industry's best steel today at temperatures below 300° F. Going back to figure 2 it is seen that no temperature has been indicated except in the known area. Temperature has been omitted, since for aircraft applications it is a variable on this chart, and increases to about 800° F as the operating conditions approach the upper right-hand corner of the chart. This means that there is much less known than is indicated by this plot, since it is true only for temperatures under 300° F - at high temperatures even the known region disappears.

The bearing life must be increased many thousandfold to meet the future objective requirements. Even stacked bearings offer only a limited solution in this regard. During the past, the life ratings of bearings have been continually improved; however, what is now asked of the rolling contact bearing is almost in the realm of fancy.

In conclusion, it is obvious that much research and development data are required on high-speed rolling-contact bearings, particularly at high temperatures.

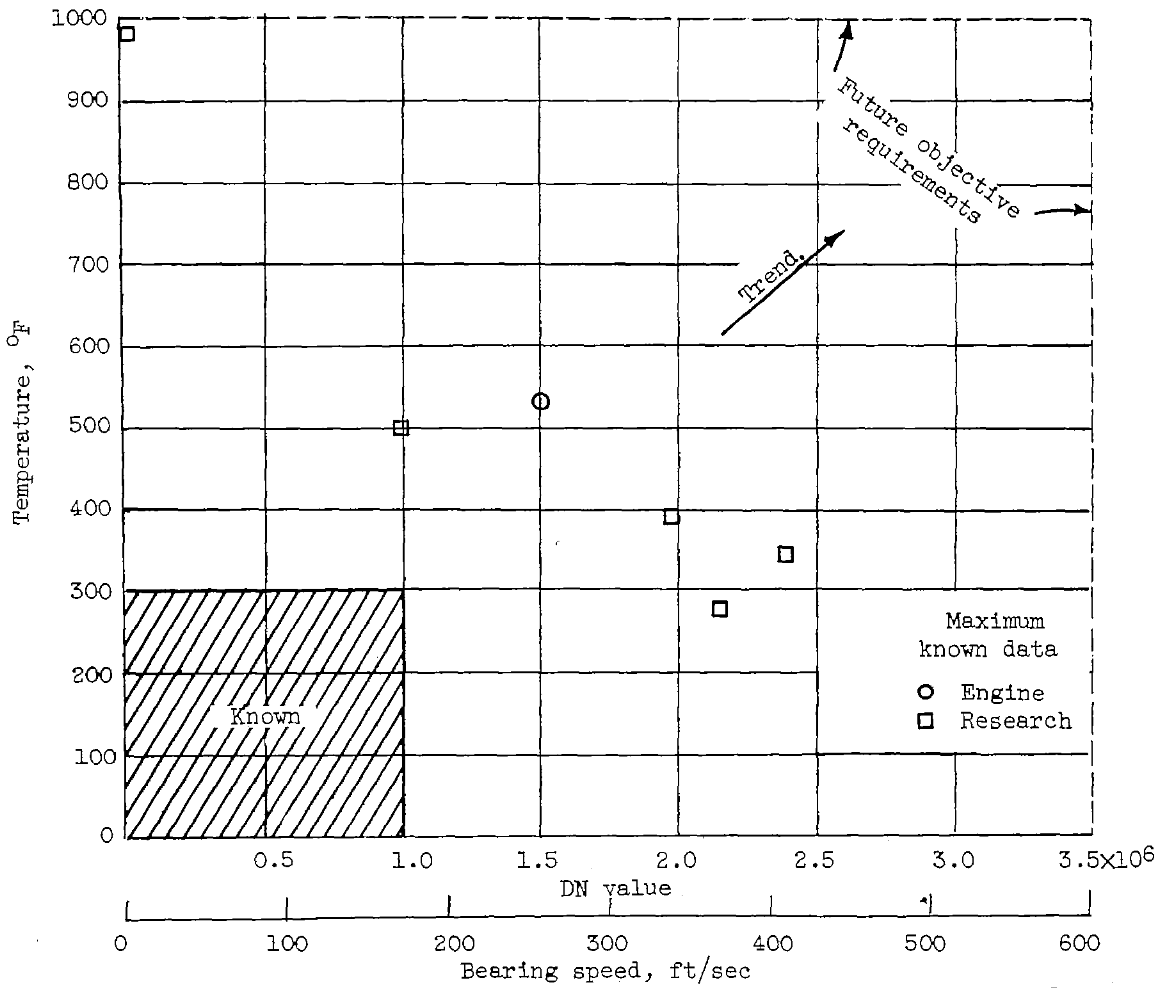

Figure 1. - High-speed, high-temperature radial load bearings. (Cylindrical roller bearings; deep groove ball bearings.)

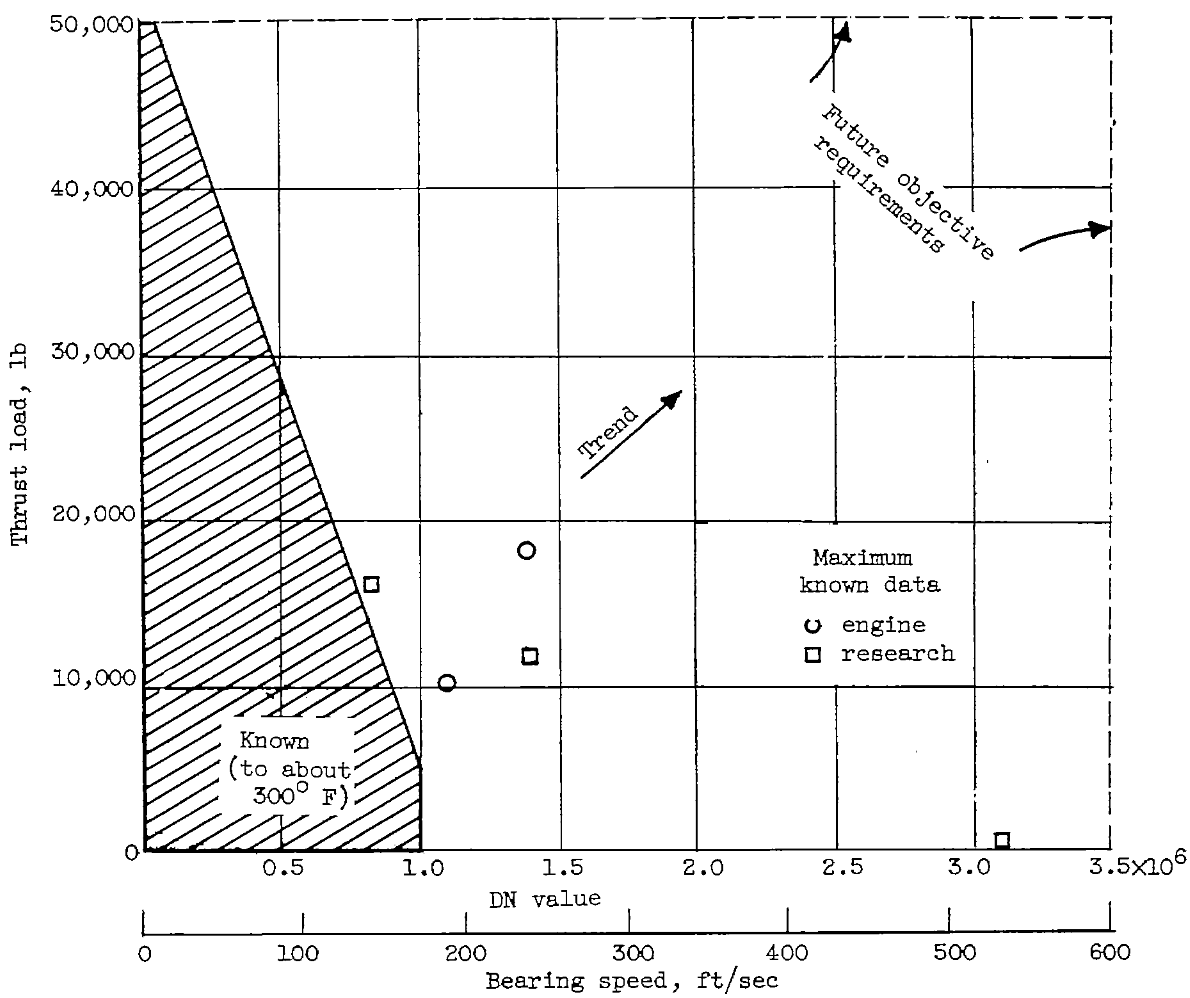

Figure 2. - High-speed, high-thrust load. Angular-contact ball bearings.

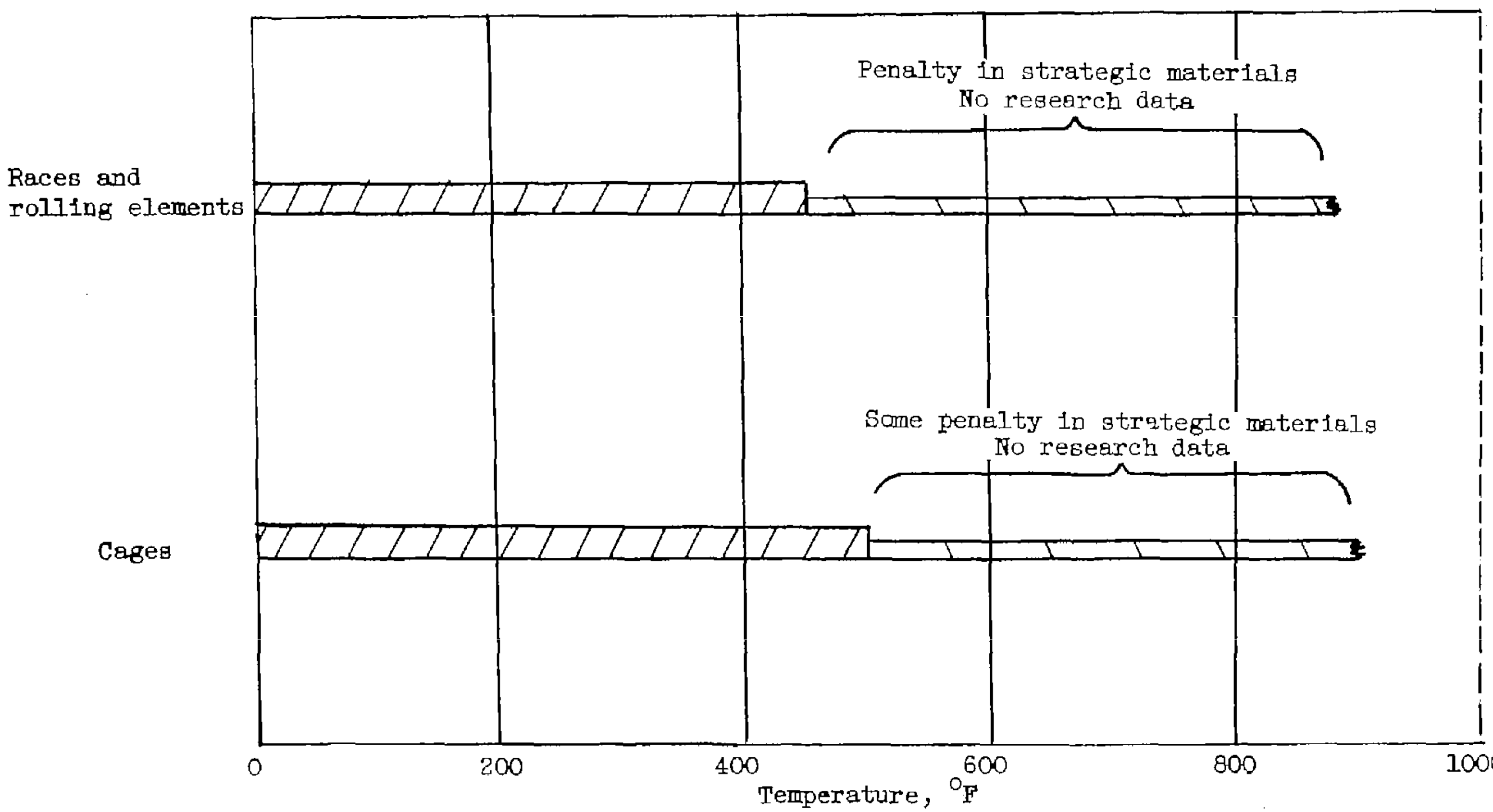

Figure 3. - Rolling contact bearing materials.

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